# USAAVLABS TECHNICAL REPORT 66-85

# ADVANCEMENT OF SPUR GEAR DESIGN TECHNOLOGY

Final Report

By

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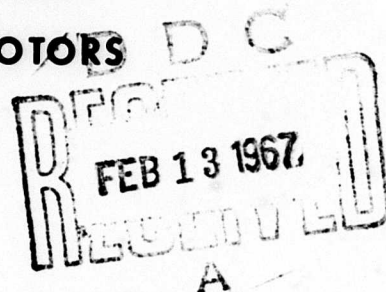December 1966

# U. S. ARMY AVIATION MATERIEL LABORATORIES
# FORT EUSTIS, VIRGINIA

## CONTRACT DA 44-177-AMC-318(T)

## ALLISON DIVISION - GENERAL MOTORS
## INDIANAPOLIS, INDIANA

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The objective of this program was to conduct an analytical and experimental study to derive or establish accurate fac·· tors for inclusion in spur gear design formula for the accu·· rate appraisal of gear bending strength.

This report presents the results of this investigation. An accurate spur gear bending strength formula was determined and an IBM 7090 computer program using the substantiated for- mula was provided.

This command concurs in the conclusions made by the contractor.

ADVANCEMENT OF SPUR GEAR DESIGN TECHNOLOGY

Final Report

EDR 4743

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W. L. McIntire and R. C. Malott

Prepared by

Allison Division ● General Motors
Indianapolis, Indiana

for

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FORT EUSTIS, VIRGINIA

# FOREWORD

**This is the final report** on the Allison project entitled "Advancement of Spur Gear **Design** Technology." This project was conducted during the 13-month period from 29 **June 1965** through 28 July 1966 for the U.S. Army Aviation Materiel Laboratories (USAAVLABS) under contract DA 44-177-AMC-318(T).

USAAVLABS technical direction was provided by Mr. R. Givens. Mr. W. L. McIntire served as the Allison project engineer. The principal investigators at Allison were Mr. R. C. Malott, Mr. F. G. Leland, Mr. K. V. Young, and Mr. W. W. Gunkel. The project was reviewed periodically with Mr. R. L. Mattson of General Motors Research for suggestions and comments.

Permission was obtained from the American Gear Manufacturers Association (AGMA) to print AGMA 220.02, Tentative AGMA Standard for Rating the Strength of Spur Gear Teeth, in this final report.

## SUMMARY

This report presents the results of an analytical and experimental program to derive and substantiate a bending strength design formula for spur gears. The program consisted of:

● Static single tooth fatigue testing of 16 gear designs in a design experiment to determine the effect of four geometric variables—diametral pitch, pressure angle, fillet size, and fillet configuration (full form ground or protuberant hobbed).

● Evaluation of the ability of five current calculation methods—AGMA, Dolan-Broghamer, Heywood, Kelley-Pedersen, and Lewis—to predict the relative ranking of the 16 fatigue test gear endurance limits.

● Statistical analyses of the fatigue test data to develop a predictive formula and relative significance values of the four geometric variables and their two- and three-factor interactions.

● A strain gage and photostress experimental evaluation to measure stress on eight of the fatigue test gears for comparison with calculated stresses and fatigue test endurance limits.

● R. R. Moore rotating beam fatigue tests of the gear material to establish basic material strength for comparison with fatigue test endurance limits.

● Measurement of the fatigue test gear crack location for comparison with location of the weakest section as predicted by the Lewis and Dolan-Broghamer calculation methods.

● Metallurgical examination of five representative fatigue test gears to verify material processing and mode of failure.

● A dynamic test at high pitch line velocities—up to 26,000 feet per minute—to determine speed effect on gear tooth bending stress.

● Development of a computer program to calculate gear tooth bending stress from the basic gear geometry, thus eliminating the need for a gear tooth layout.

The results of the program were as follows:

● The AGMA method of calculating gear tooth bending stress predicted the greatest number of correct rankings of the 16 fatigue test gear endurance limits. This method also predicted the rank position with the least average error.

● Comparison of endurance limits, based on applied load, calculated from the fatigue test data for each of the 16 gear designs was made by statistical tests of significance. Diametral pitch and pressure angle had a significant effect on gear tooth bending fatigue strength. The AGMA formula successfully compensated for the significant variables determined by the base-line applied load analyses.

- The strain gage stress values obtained tend to verify the AGMA calculated stresses. The average strain rate measured on the fatigue test gears was within 2.5 percent of the strain rate calculated by the AGMA formula.
- The basic gear material endurance limit determined by the R. R. Moore rotating beam test was 182,000 p.s.i. when modified for single-direction bending. The fatigue test gear average endurance limit based on AGMA calculated stress was 182,000 p.s.i. It appears, therefore, that basic material strength can be very closely related to AGMA calculated gear stress and endurance limit.
- Fatigue test gear crack location was nearer the Dolan-Broghamer than the Lewis predicted location, as expected.
- Metallurgical examinations verified good processing of the fatigue test gears and fatigue as the mode of failure. Failures were initiated at random locations across the face width of the gears, indicating minimal influence of surface finish, material inclusions, corner edge break, and test rig alignment.
- Steady hoop stresses were measured in the dynamic test at the weakest section. The measured stresses were 70 percent of the calculated root diameter hoop stress. The measured stress was 14,000 p.s.i. which is considered sufficient to necessitate its inclusion in bending stress determinations for high-speed gears.
- The dynamic test also measured dynamic fluctuating gear tooth level stresses. Stresses indicated a dynamic stress factor increasing with the square of the rotational speed. The dynamic factor was 1.8 at 26,000-feet-per-minute-pitch-line velocity.
- The computer program developed accurately determined the root fillet configuration by calculating the true radius or trochoidal fillet depending on the manufacturing method and the tool (hob) dimensions. The Lewis weakest section is determined by iteration. The gear tooth dimensions determined are used in the AGMA formula to determine bending stress. A hoop stress at the root diameter is then calculated to account for the effect of speed on gear tooth bending stress. The steady hoop stress and the fluctuating bending stresses are then combined by means of a modified Goodman diagram to produce a combined stress and an expected failure life. The modified Goodman diagram was based on the average S/N curve determined by the fatigue test gears.

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# INTRODUCTION

The purpose of the project was to conduct an analytical and experimental investigation to derive factors and formulae which can be used to appraise accurately spur gear tooth bending strength for aircraft applications.

The objective of the project was twofold—to substantiate an accurate spur gear bending strength formula and to provide an IBM 7090 computer program using the substantiated formula. Correlation of a basic material strength with this formula was desired.

There are four common modes of gear failure—tooth breakage, surface pitting, scoring, and wear. Tooth breakage is the most severe and often causes considerable secondary damage and sometimes catastrophic failure of an entire gear unit. It may be caused accidentally, such as when a foreign object passes through a tooth mesh, or it may be caused by the repetitive high bending stresses near the root of the tooth when under load.

Many factors affecting the bending fatigue strength of gear teeth are not treated with precision in current spur gear design formulae. This is because the magnitude and interrelationships of the various factors have not been accurately assessed. Gear tooth bending strength is a function of geometric variables such as pressure angle, diametral pitch, tooth width, root fillet form, and root fillet radius. It is also influenced by manufacturing variables such as surface finish, residual stress, material, and processing technique. Operating variables such as speed, alignment, dynamic loading, and vibration affect the fatigue life. A thorough analysis of these variables will permit more accurate assessment of gear life expectancy.

Considerable research has been accomplished in analyzing gear tooth bending strength; however, there is wide variation in the type of analysis, test data, and field experience. In many instances extensive extrapolation has been required to apply these data to carburized gears designed to current standard geometric proportions. The program described herein was conducted in an effort to establish correlation between analytical methods and actual test results for lightweight aircraft gearing.

Current methods of calculating gear tooth bending stress are based on analytical studies and photoelastic tests. These methods produce calculated stresses which are appreciably lower than measured gear stresses and basic material strengths. Thus the calculations are most often used to compare similar designs. An "ideal" gear tooth bending strength formula would relate the operating gear tooth stress to the basic material strength in such a way as to produce a gear life which has been substantiated by fatigue test. It was therefore the intent of the subject program to provide a more accurate bending stress formula by also relating calculated stress and fatigue test results to the basic material strength. R. R. Moore tests of carburized specimens were used to provide a basic material strength.

To accomplish the program, the following analytical and experimental analyses were conducted.

- Design Analysis—An analytical review was made of current spur gear tooth bending strength formulae. Each formula was analyzed and compared to determine the effects of design variables.
- Experimental Evaluation—A photostress analysis was conducted to evaluate the location and distribution of the maximum stress on actual fatigue test gears. Strain gage stress measurements were obtained for correlation with stress calculations.

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- Gear Tooth Fatigue Tests—A single tooth fatigue test was conducted to investigate the effect of diametral pitch, pressure angle, root fillet size, and root fillet configuration on fatigue life. Eighty gears were manufactured. Extreme care was taken to reduce all possible manufacturing variances which might affect fatigue life. Metallurgical investigations of the fatigue failures were also made to ensure that the basic material was sound and was properly heat treated. Four teeth on each gear were available for fatigue testing.
- R. R. Moore Tests—R. R. Moore tests were conducted using the same heat of material used for the test gears. The data obtained were used for comparison with the bending endurance strengths from the gear fatigue tests.
- Dynamic Tests—An existing accessory gear in an Allison 501-D13 gearbox was instrumented with strain gages. The gear was operated at high speed (pitch line velocity of 27,000 feet/minute) at load and no-load conditions to investigate the effect of speed on bending stress. The data obtained were reduced to determine the effect of centrifugal and dynamic loads on bending stress.
- Final Computer Program—Data from the previously mentioned items were formulated into an IBM 7090 computer program for spur gear bending strength.

## ANALYSIS OF PROBLEM

### HISTORICAL REVIEW

A review of gear tooth bending strength theory was made. The results of this review are discussed in the following paragraphs.

In 1887, Mr. A. B. Couch in an American Society of Mechanical Engineers (ASME) meeting was asked for a rule to determine safe gear loads (reference 62). He expressed surprise and replied that "the rules furnished (available) are in number bountiful and in variety nearly infinite." He reported that a fellow ASME member had compiled a list of 30 to 40 such rules. In these different rules, safe load varied directly as the square and in a few instances even as the cube of circular pitch. Face width was the only other widely considered factor. The same discussion group expressed an awareness of dynamic loads when they commented, "The cog gearing of power levers used in threshing, owing to the irregular draft of horses, is subjected to heavier strains."

In 1892, Mr. Wilfred Lewis presented a paper which related gear tooth bending strength to tooth geometry. The formula derived in this paper is the basis for most bending stress calculation methods used today. Publication of the Lewis formula did not result in its immediate unanimous adoption. However, it did accelerate further analytical and experimental investigations. Charts and computer programs based on the Lewis formula were developed to expedite gear designs (references 27 and 44). A cantilever beam bending formula for a rectangular section was used to calculate bending stress from 100-times size gear tooth layouts at successive sections 0.100-inch apart to determine the minimum load section for an arbitrary constant stress (reference 31). This work served to verify the principles of the Lewis formula. The improved accuracy required and the higher peripheral speeds of gears necessitated three basic changes to the Lewis formula which have been accepted by general usage—the addition of the Dolan-Broghamer stress concentration factor, the addition of a compressive stress term, and consideration of tooth loading at the high point of single tooth contact or at the pitch diameter rather than at the tip.

The Dolan-Broghamer stress concentration formula is based on photoelastic stress work accomplished at the University of Illinois Engineering Experiment Station in 1942 (reference 16). Their formula is included in the current AGMA Standard 220:02 which is included in this report as Appendix VI. This formula is included in many stress and engineering handbooks as a modified Lewis formula or as a part of the AGMA standard.

Other investigators have obtained photoelastic stress results in close agreement with those of Dolan and Broghamer (references 1 and 10). Prior to the Dolan-Broghamer formula, the stress concentration factors included only a limited number of geometric variables and thus were not as universally applicable (reference 58).

The existence of stresses other than bending stresses in the critical root area of a gear tooth was recognized at an early date. Calculation and vectorial addition of shear stress, from the tangential (circumferential) component of the tooth load, were accomplished and published in 1897 (reference 31). Several current tooth strength formulae include shear stress; the AGMA standard does not. See Appendix VI. For a given tooth load, shear stress would be greate. a pressure angle gear of 14.5 degrees than for a similar one of 25 degrees.

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Compressive stress from the tooth load radial component has been accepted for summation with the gear tooth bending stress. The AGMA standard (Appendix VI) includes a compressive stress term. More recently, an additional compressive stress at the tensile root fillet has been expressed. This additional stress is due to the moment about the gear tooth radial center line from the radial component of the tooth load. An unsymmetrical stress distribution across the weakest section results, which tends to relieve the bending stresses in both the tensile (load side) and compressive (unloaded side) root fillet areas. The gear tooth load components are shown in Figure 1. These static stresses are present in the photoelastic models used to determine stress concentration factors. Thus, their effect is included in the stress concentration factor if the calculated stress used as a basis does not include any such component load stress.
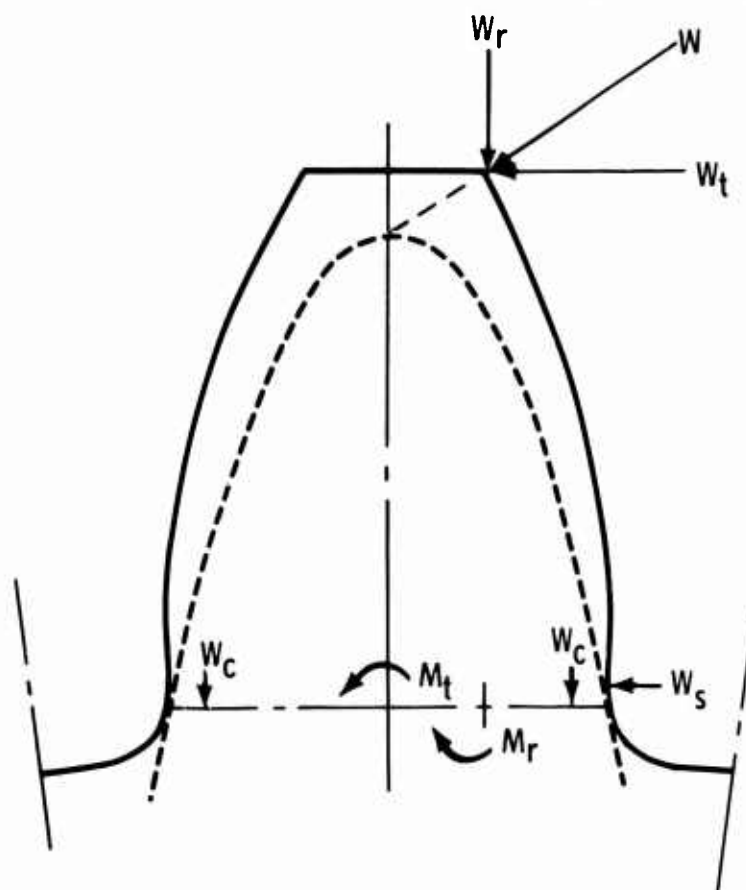
W —normal applied load

$W_t$ —tangential component of W

$W_r$ —radial component of W

$W_c$ —compressive load at weakest section from $W_r$

$W_s$ —shear load at weakest section from $W_t$

$M_t$ —bending moment at weakest section from $W_t$

$M_r$ —bending moment at weakest section from $W_r$

Figure 1. Gear Tooth Static Load Analysis.

Tip loading, as used in the original Lewis formula, was often changed to pitch line loading to account for load sharing at the tip. It was only recently that the exact point of maximum loading for spur gears was recognized (reference 61). This latest refinement permitted more accurate assessment of safety and/or dynamic factors.

Speed effect curves were developed from experimental data on cast iron gears which had been operated under increasing load until tooth breakage occurred (reference 42). The shape of the curves was similar to the curves currently in the AGMA standard (speed effect becomes constant at higher speeds). The same curve shape can also be observed in current gear scoring versus speed work curves (reference 8).

A review of the Engineering Index volumes for 1950 through 1965 reveals approximately 1255 abstracts on gears. Ten percent of these involve gear tooth bending strength calculation, fatigue testing, or dynamic factors. Almost 20 percent are from foreign sources, mostly German. The yearly output of such articles is nearly constant over this time period.

Several gear tooth strength formulas are of current interest. Five have been investigated and applied to the 16 fatigue test gear configurations—Lewis, Dolan-Broghamer, Heywood, Kelley-Pedersen, and AGMA. A full ground root fillet radius was assumed for all gears in this study. The stresses for each configuration are listed in Table I. The average, range, and variation in stress for each method relative to the Lewis stress are shown in Figure 2. The Kelley-Pedersen method produced a high average stress and by far the greatest range of stress (75 percent of the average Lewis stress). The average stress of the 16 gears as computed by the five formulas varied from 150 to 187 percent of the average Lewis stress. The AGMA method produced the smallest average stress and the smallest range (20 percent of the average Lewis stress). In contrast, the Lewis stresses calculated for the 16 test gear configurations loaded to 1000 pounds per inch of face width varied by over 400 percent. All five formulas identify the same configurations as having the highest and the lowest stresses (boxed numbers in Table I). The highest stresses are most often calculated by the Heywood method, while the lowest stresses in all cases were determined by the Lewis formula, which does not consider stress concentration.

The geometric construction and formula for each of the five gear tooth strength calculation methods are shown in Figures 3, 4, and 5 and in Tables II and III. The Dolan-Broghamer and AGMA methods use Lewis geometric construction (Figure 3) and thus are similar to each other. A detailed discussion of the Dolan-Broghamer and AGMA methods and factors is given in the section titled Discussion of Results.

The Heywood and Kelley-Pedersen construction methods (Figures 4 and 5, respectively) incorporate features which generally lower the position of the weakest section. The Heywood construction method contains several arbitrary features which are not suitable for use with all gear design systems. Variations such as nonstandard addendums and dedendums, which are often used in aircraft designs to balance bending strength or sliding velocity, are examples.

The Kelley-Pedersen method constructs the Lewis parabola, then rotates the tangent line around the root fillet through a "stress shift" angle. Both the Kelley-Pedersen and Heywood methods contain stress concentration factor terms.

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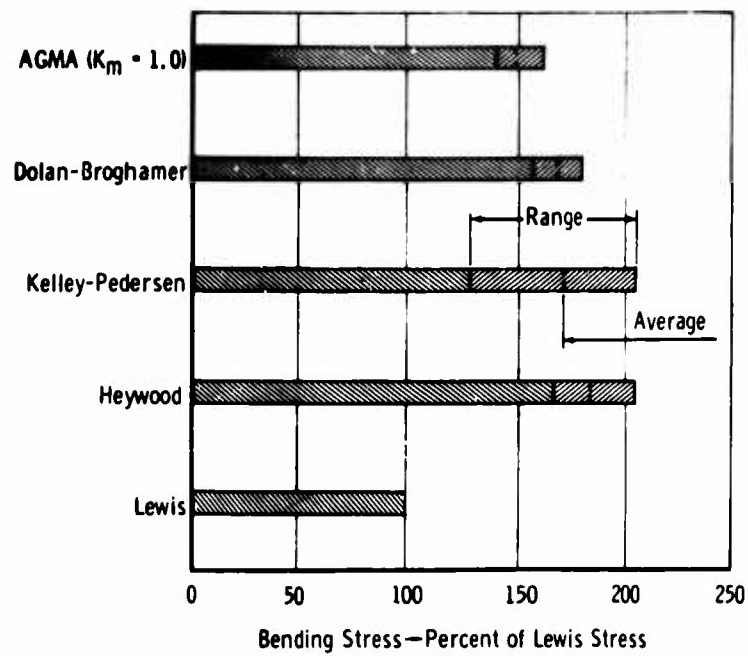
Figure 2.  Relative Gear Tooth Bending Stress.



$$S_b = \frac{W}{F\,Y\,p}$$

$$Y = \frac{2}{3}\frac{x}{p}$$

$$x = \frac{t^2}{4h}$$

$$S_b = \frac{6\,Wh}{Ft^2}$$
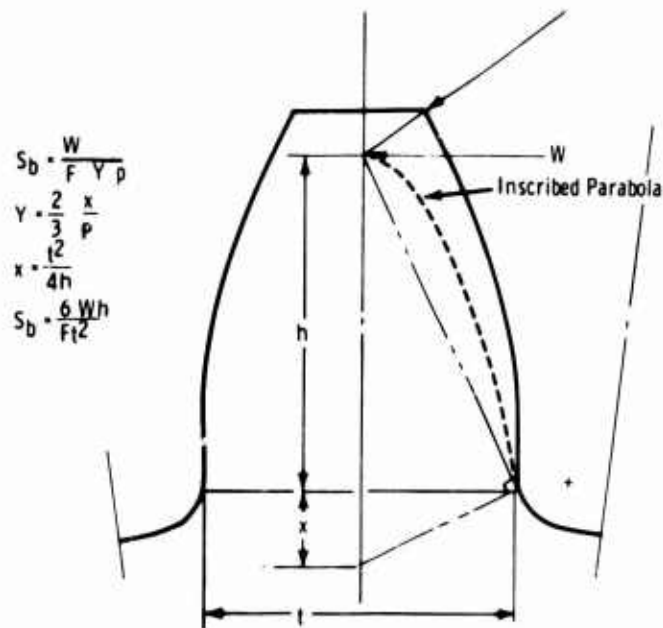
where :

W  •  tangential component of load applied at vertex of inscribed parabola

F  •  face width of tooth

$S_b$  •  maximum bending stress

h  •  height of equivalent constant stress parabolic beam

t  •  thickness of beam at weakest section

p  •  circular pitch

Figure 3.  Lewis Construction and Gear Tooth Bending Stress Formula.

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## TABLE I
## COMPARISON OF GEAR TOOTH BENDING STRESSES
## CALCULATED BY VARIOUS METHODS

| Gear Configuration | | | | | Gear Tooth Str‹ | | | |
|---|---|---|---|---|---|---|---|---|
| Gear | Pitch | Pressure Angle (deg) | Radius (in.) | Unit Load (lb) | Lewis | Dolan-Broghamer | Dolan-Broghamer | A‹ |
| 1 | 6 | 20 | 0.050 | 6,000 | 12,692 | 22,682 | 179 xx | 2 ‹ |
| 3 | 6 | 20 | 0.080 | 6,000 | 11,020 | 19,382 | 176 | 1 ‹ |
| 5 | 6 | 20 | 0.050* | 6,000 | 17,572 | 28,385 | 162 | 2 ‹ |
| 7 | 6 | 20 | 0.080* | 6,000 | 14,023 | 22,796 | 163 | 2 ‹ |
| 9 | 6 | 25 | 0.050 | 6,000 | 9,871 | 17,583 | 178 | 1 ‹ |
| 11 | 6 | 25 | 0.067 | 6,000 | [9,447] | [16,651] | 176 | [1 ‹ |
| 13 | 6 | 25 | 0.050* | 6,000 | 11,028 | 18,673 | 169 | 1 ‹ |
| 15 | 6 | 25 | 0.067* | 6,000 | 10,468 | 17,574 | 168 | 1 ‹ |
| 2 | 12 | 20 | 0.025 | 12,000 | 27,391 | 47,781 | 174 | 4 ‹ |
| 4 | 12 | 20 | 0.040 | 12,000 | 23,869 | 40,944 | 171 | 3 ‹ |
| 6 | 12 | 20 | 0.025* | 12,000 | [38,497] | [60,520] | 157 x | [5 ‹ |
| 8 | 12 | 20 | 0.040* | 12,000 | 30,687 | 48,562 | 158 | 4 ‹ |
| 10 | 12 | 25 | 0.025 | 12,000 | 21,159 | 36,732 | 174 | 3 |
| 12 | 12 | 25 | 0.033 | 12,000 | 20,306 | 34,893 | 172 | 2 ‹ |
| 14 | 12 | 25 | 0.025* | 12,000 | 23,630 | 39,044 | 165 | 3 |
| 16 | 12 | 25 | 0.033* | 12,000 | 22,448 | 36,806 | 164 | 3 ‹ |
| Average | | | | | 19,007 | 31,813 | 167.4 | 2 ‹ |
| Variation (M. ÷ Min) | | | | | 4.075 | 3.635 | 1.140 | ‹ |

\* Root diameter for protuberance cut.
x designates low stress range configuration.
xx designates high stress range configuration.

Notes:
    A value of 1.0 was used for $K_m$ (load distribution factor).
    High and low calculated stress configurations are boxed.

## Tooth Stress at High Point of Single Tooth Contact (p. s. i. )

| mer | AGMA | AGMA as % of Lewis | Heywood | Heywood as % of Lewis | Kelley-Pedersen | Kelley-Pedersen % of Lewis |
|---|---|---|---|---|---|---|
| | 20,484 | 161 xx | 24,504 | 193 | 24,229 | 191 |
| | 17,300 | 157 | 19,750 | 179 | 19,654 | 178 |
| | 26,152 | 149 | 31,266 | 178 | 27,770 | 158 |
| | 20,729 | 148 | 23,614 | 168 | 19,518 | 139 |
| | 14,952 | 151 | 20,279 | 205 xx | 20,305 | 206 xx |
| | 14,063 | 149 | 18,093 | 192 | 17,512 | 185 |
| | 16,148 | 146 | 21,900 | 199 | 21,767 | 197 |
| | 15,099 | 144 | 19,398 | 185 | 18,619 | 178 |
| | 43,006 | 157 | 51,737 | 189 | 51,859 | 189 |
| | 36,447 | 153 | 41,710 | 175 | 41,848 | 175 |
| | 55,548 | 144 | 67,120 | 174 | 57,038 | 148 |
| | 44,015 | 143 | 50,531 | 165 x | 39,402 | 128 x |
| | 31,196 | 147 | 42,527 | 201 | 40,272 | 190 |
| | 29,456 | 145 | 38,093 | 188 | 34,754 | 171 |
| | 33,680 | 143 | 45,997 | 195 | 43,453 | 184 |
| | 31,562 | 141 x | 40,888 | 182 | 37,195 | 166 |
| | 28,115 | 147.9 | 34,838 | 183.3 | 33,233 | 169.4 |
| | 3.950 | 1.142 | 3.710 | 1.242 | 3.257 | 1.493 |

$$S_b = \left[1 + 0.26\left(\frac{e}{R}\right)^{0.7}\right]\left[\frac{1.5\,a}{e^2} + \left(\frac{0.36}{be}\right)^{0.5}(1 + 0.25\sin\gamma)\right]\frac{P}{F}$$

where:

$h_t$ = tooth depth

$F$ = face width

$a$ = moment arm

$e$ = resisting material

$R$ = fillet radius

$P$ = normal load

$\gamma$ = angle deviation of load from the point of loading to the point of maximum stress

$b$ = distance parallel to equivalent straight-sided projection from the point of loading to the point of maximum stress

$S_b$ = maximum fillet stress
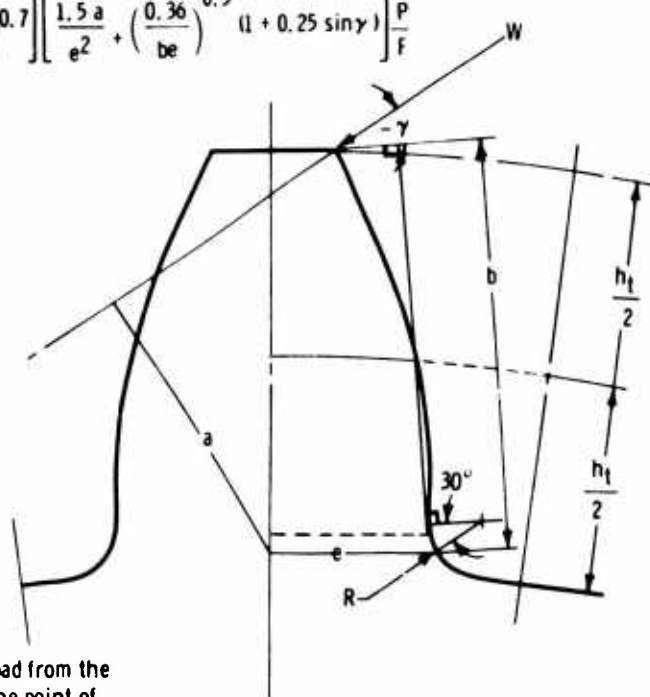
Figure 4. Heywood Construction and Gear Tooth Bending Stress Formula.

$$S_b = \frac{W}{F}\left[1 + 0.26\left(\frac{e}{r_f}\right)^{0.7}\right]\left[\frac{1.5\,a}{e^2} + \frac{\sin\beta}{2\,e} + \frac{0.45}{(be)^{0.5}}\right]$$

where:

$S_b$ = maximum fillet stress

$W$ = normal load

$F$ = tooth face width

$e$ = dimension of resisting material

$r_f$ = fillet radius at the point of maximum stress

$b$ = distance from point of load application to maximum stress

$\beta$ = deviation of load line from direction of principal stress

Lewis Inscribed Parabola

$\epsilon = 25^\circ \dfrac{a}{2}$
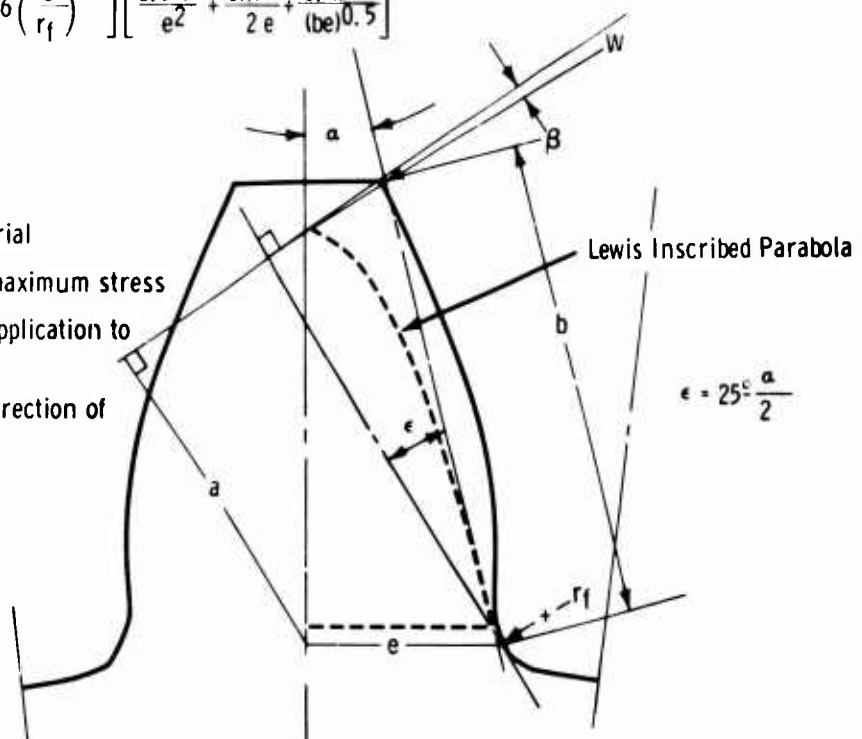
Figure 5. Kelley-Pedersen Construction and Gear Tooth Bending Stress Formula.

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TABLE II
## DOLAN-BROGHAMER GEAR TOOTH BENDING STRESS FORMULA

$$S_b = K \left[ \frac{6 \, Wh}{t^2 F} - \frac{W}{tF} \tan \phi_L \right]$$

where

| | | |
|---|---|---|
| $W$ | $=$ | tangential load at load point |
| $\phi_L$ | $=$ | pressure angle at load point |
| $h, \, t$ | $=$ | load height and maximum stress section tooth thickness from gear tooth layout (Lewis construction) |
| $F$ | $=$ | gear tooth face width |
| $S_b$ | $=$ | combined stress (from radial and flexural components of load) at the tensile fillet |
| $K$ | $=$ | concentration factor for combined stress at tensile fillet |
| | $=$ | $\dfrac{\text{maximum observed tensile stress}}{\text{computed combined stress}}$ |
| | $=$ | $0.22 + \left(\dfrac{t}{r_f}\right)^{0.2} \left(\dfrac{t}{h}\right)^{0.4}$ for 14.5-degree pressure angle |
| | $=$ | $0.18 + \left(\dfrac{t}{r_f}\right)^{0.15} \left(\dfrac{t}{h}\right)^{0.45}$ for 20-degree pressure angle |
| $r_f$ | $=$ | minimum fillet radius at bottom of the trochoidal fillet of a generated tooth as determined by procedure developed by Mr. A. H. Candee. |
| | $=$ | $r_i + r_t$ |
| $r_i$ | $=$ | $b_i{}^2/(R = b_i)$ = minimum radius of curvature of trochoid at center of edge radius |
| $b_i$ | $=$ | $b - r_t$ = dedendum to center of tool edge radius |
| $r_t$ | $=$ | tool edge radius |
| $b$ | $=$ | length of dedendum of the gear |
| $R$ | $=$ | radius of the pitch circle |
| $t$ | $=$ | thickness of tooth at theoretical weakest section (Lewis) |
| $h$ | $=$ | height of load position above the theoretical weakest section |

TABLE III
## AGMA GEAR TOOTH BENDING STRESS FORMULA

$$S_t = \frac{W_t K_o}{K_v} \left(\frac{P_d}{F}\right) \frac{K_s \, K_m}{J}$$

where

| | | | |
|---|---|---|---|
| $S_t$ | $=$ | calculated tensile stress at the root of the tooth | |
| $W_t$ | $=$ | transmitted tangential load at operating pitch diameter | |
| $K_o$ | $=$ | overload factor | Load |
| $K_v$ | $=$ | dynamic factor | |
| $P_d$ | $=$ | transverse diametral pitch | |
| $F$ | $=$ | net face width | Tooth Size |

TABLE III (CONT)
AGMA GEAR TOOTH BENDING STRESS FORMULA

$K_s$ = size factor

$K_m$ = load distribution factor $\Big\}$ Stress Distribution

$J$ = geometry factor

$J$ = $\dfrac{Y}{K_f\, m_N}$ for spur gears

$Y$ = tooth form factor

$K_f$ = stress correction factor

$m_N$ = load sharing ratio

$K_f$ = $H + \left(\dfrac{t}{r_f}\right)^J \left(\dfrac{t}{h}\right)^L$ = Dolan-Broghamer Stress Concentration Factor

| | | | Pressure Angle (Degrees) |
|---|---|---|---|
| H | = | 0.22 | 14.5 |
| | | 0.18 | 20 |
| J | = | 0.20 | 14.5 |
| | | 0.15 | 20 |
| L | = | 0.40 | 14.5 |
| | | 0.45 | 20 |

$t$, $h$, and $r_f$ from gear tooth layout (Lewis construction)

$m_N$ = normally 1 for spur gears

$Y$ = $\dfrac{1}{\dfrac{\cos\phi_L}{\cos\phi}\left(\dfrac{1.5}{X} - \dfrac{\tan\phi_L}{t}\right)}$ for spur gears

$\phi$ = tooth pressure angle

$\phi_L$ = load pressure angle

$t$ = tooth thickness at the section of maximum stress (Lewis construction)

$X$ = tooth strength factor from layout (Lewis construction)

$r_f$ = radius of curvature of fillet at point tangent to root circle (may also be calculated)

$S_t \lesseqgtr \dfrac{S_a\, K_L}{K_T\, K_R}$

where

$S_a$ = allowable stress for material

$K_L$ = life factor

$K_T$ = temperature factor

$K_R$ = factor of safety

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In summary, review of the literature indicated that wide variations of bending strength could be calculated for a given configuration. Little data are available which attempt to correlate basic material strengths from laboratory tests with actual gears. It was thus apparent that a controlled fatigue experiment with full-size tooth proportions could aid the development of a more accurate method of calculating bending strength. Basic material strength data from R. R. Moore tests for correlation would also enhance the analysis.

## DESIGN OF EXPERIMENT

Four factors of gear tooth geometry were investigated in a statistically designed experiment. Each of the factors selected was expected to affect gear tooth life. The experiment was designed to indicate if these factors interacted and if the observed results were statistically significant. The geometric factors evaluated were:

| Factor | Levels | Values assigned |
|---|---|---|
| ● Diametral pitch | 2 | 6 and 12 |
| ● Pressure angle | 2 | 20 and 25 degrees |
| ● Root radius size | 2 | Small and large (exact values dependent on diametral pitch) |
| ● Fillet configuration | 2 | Full form ground and protuberance hobbed |

The experiment planned involved cycling three gear teeth to failure at each of four stress levels for each of the 16 possible combinations of the four geometric factors investigated. Evaluation of the effects of the four geometric factors was to be based on the finite life portion of the resulting fatigue (S/N) curves.

## DESIGN OF FATIGUE TEST GEARS

Drawings of the 16 fatigue test gears are presented in Appendix I. Table IV lists the pertinent dimensions for the 16 fatigue test gear configurations.

Diametral pitch values of 6 and 12 were selected. A diametral pitch of 6 is typical for main power train gears in turboprop and helicopter aircraft engine transmissions. A diametral pitch of 12 provides a reasonable 2:1 variation; it also represents typical aircraft engine accessory drive train practice.

The pressure angles of 20 and 25 degrees were selected since they represent aircraft engine design practice.

Each gear tooth design has a maximum fillet radius size that can be accommodated between the active profile diameter and the root diameter. Using this maximum value of 100 percent, the minimum fillet radii for the test gears were specified as 80 percent for one design experiment level. The other level was set at 50 percent for the 20-degree pressure angle gears and 60 percent for the 25-degree gears to maintain a minimum actual fillet radius of 0.025 inch. A manufacturing tolerance of 20 percent was thus provided with a minimum variation of 20 percent in fillet size.

The fatigue test gears were made without a rim and web to eliminate possible complications. Twenty-four tooth gears were chosen to avoid undercutting and to provide reasonable gear sizes.

TABLE IV
FATIGUE TEST GEAR DIMENSIONS

| Configuration | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | |
|---|---|---|---|---|---|---|---|---|---|
| Part number | EX-78772 | EX-78773 | EX-78774 | EX-78775 | EX-78776 | EX-78777 | EX-78778 | EX-78779 | EX-7. |
| Number of teeth | 24 | 24 | 24 | 24 | 24 | 24 | 24 | 24 | 24 |
| Pressure angle, degrees | 20 | 20 | 20 | 20 | 20 | 20 | 20 | 20 | 25 |
| Diametral pitch | 6 | 12 | 6 | 12 | 6 | 12 | 6 | 12 | 6 |
| Pitch diameter, inches | 4.0 | 2.0 | 4.0 | 2.0 | 4.0 | 2.0 | 4.0 | 2.0 | 4.0 |
| Base circle diameter, inches | 3.7588 | 1.8794 | 3.7588 | 1.8794 | 3.7588 | 1.8794 | 3.7588 | 1.8794 | 3.62 |
| Diameter at HPSTC*, inches | 4.08289 | 2.04748 | 4.08289 | 2.04748 | 4.08289 | 2.04748 | 4.08289 | 2.04748 | 4.13 |
| Active profile diameter, inches | 3.7984 | 1.8969 | 3.7984 | 1.8969 | 3.7984 | 1.8969 | 3.7984 | 1.8969 | 3.75 |
| Addendum factor | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 |
| Dedendum factor | 1.25 | 1.25 | 1.25 | 1.25 | 1.40 | 1.40 | 1.40 | 1.40 | 1.20 |
| Whole depth factor | 2.25 | 2.25 | 2.25 | 2.25 | 2.40 | 2.40 | 2.40 | 2.40 | 2.20 |
| Outside diameter, inches | 4.333 | 2.167 | 4.333 | 2.167 | 4.333 | 2.167 | 4.333 | 2.167 | 4.33 |
| Root diameter, inches | 3.583 | 1.792 | 3.583 | 1.792 | 3.533 | 1.767 | 3.533 | 1.767 | 3.60 |
| Minimum fillet radius, inches | 0.050 | 0.025 | 0.080 | 0.040 | 0.050 | 0.025 | 0.080 | 0.040 | 0.05 |
| Maximum possible fillet radius, inches | 0.1008 | 0.0506 | 0.1008 | 0.0506 | 0.1008 | 0.0506 | 0.1008 | 0.0506 | 0.08 |
| Minimum fillet radius**, percent | 50 | 50 | 80 | 80 | 50 | 50 | 80 | 80 | 60 |
| Fillet type | ←——— | Full Ground | ——————→ | | ←——— | Protuberant | ——————→ | | ←— |
| Tooth thickness, inches | 0.2618 0.2598 | 0.1309 0.1289 | 0.2618 0.2598 | 0.1309 0.1289 | 0.2618 0.2598 | 0.1309 0.1289 | 0.2618 0.2598 | 0.1309 0.1289 | 0.26 0.25 |
| Face width, inches (±0.002) | 0.50 | 0.25 | 0.50 | 0.25 | 0.50 | 0.25 | 0.50 | 0.25 | 0.50 |
| Contact ratio | 1.5403 | 1.4780 | 1.5403 | 1.4780 | 1.5403 | 1.4780 | 1.5403 | 1.4780 | 1.38 |

*HPSTC—high point of single tooth contact.
**Percent of maximum possible.

| | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 |
|---|---|---|---|---|---|---|---|---|---|
| ~~78 | EX-78779 | EX-78780 | EX-78781 | EX-78782 | EX-78783 | EX-78784 | EX-78785 | EX-78786 | EX-78787 |
| | 24 | 24 | 24 | 24 | 24 | 24 | 24 | 24 | 24 |
| | 20 | 25 | 25 | 25 | 25 | 25 | 25 | 25 | 25 |
| | 12 | 6 | 12 | 6 | 12 | 6 | 12 | 6 | 12 |
| | 2.0 | 4.0 | 2.0 | 4.0 | 2.0 | 4.0 | 2.0 | 4.0 | 2.0 |
| | 1.8794 | 3.6252 | 1.8126 | 3.6252 | 1.8126 | 3.6252 | 1.8126 | 3.6252 | 1.8126 |
| | 2.04748 | 4.1324 | 2.0729 | 4.1324 | 2.0729 | 4.1324 | 2.0729 | 4.1324 | 2.0729 |
| | 1.8969 | 3.7571 | 1.8759 | 3.7571 | 1.8759 | 3.7571 | 1.8759 | 3.7571 | 1.8759 |
| | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 |
| | 1.40 | 1.20 | 1.20 | 1.20 | 1.20 | 1.35 | 1.35 | 1.35 | 1.35 |
| | 2.40 | 2.20 | 2.20 | 2.20 | 2.20 | 2.35 | 2.35 | 2.35 | 2.35 |
| | 2.167 | 4.333 | 2.167 | 4.333 | 2.167 | 4.333 | 2.167 | 4.333 | 2.167 |
| | 1.767 | 3.600 | 1.800 | 3.600 | 1.800 | 3.550 | 1.775 | 3.550 | 1.775 |
| | 0.040 | 0.050 | 0.025 | 0.067 | 0.033 | 0.050 | 0.025 | 0.067 | 0.033 |
| | 0.0506 | 0.0836 | 0.0418 | 0.0836 | 0.0418 | 0.0836 | 0.0417 | 0.0836 | 0.0417 |
| | 80 | 60 | 60 | 80 | 80 | 60 | 60 | 80 | 80 |
| | → | ← Full Ground → | | | | ← Protuberant → | | | |
| | 0.1309 | 0.2618 | 0.1309 | 0.2618 | 0.1309 | 0.2618 | 0.1309 | 0.2618 | 0.1309 |
| | 0.1289 | 0.2598 | 0.1289 | 0.2598 | 0.1289 | 0.2598 | 0.1289 | 0.2598 | 0.1289 |
| | 0.25 | 0.50 | 0.25 | 0.50 | 0.25 | 0.50 | 0.25 | 0.50 | 0.25 |
| | 1.4780 | 1.3823 | 1.3230 | 1.3823 | 1.3230 | 1.3823 | 1.3240 | 1.3823 | 1.3240 |

β

Face widths of 0. 500 inch for the 6-pitch gears and 0. 250 inch for the 12-pitch gears were selected to provide slightly larger axial width than tooth thickness at the weakest section in bending. The face widths maintain proportional similarity between the two gear pitches. Carburized case depths were also varied to maintain proportional similarity.

Two root fillet configurations are in general use in aircraft gearing—full form ground and protuberance hobbed. Since almost all aircraft engine gears have ground involute profile surfaces, the root fillet radii can be ground during the same operation, thus producing a "full form" ground gear. The ground root area is subject to grinding burns, excessive case removal, and/or high residual stresses if the grinding procedures are not carefully specified and controlled. Ground root fillets may be produced by formed wheels with true radii or specially shaped fillets, or by generation which produces trochoidal fillets.

Hobbing the gear with a special hob that has protrusions at the tips results in a controlled amount of undercut in the root area, thus producing a protuberance gear. Involute grinding can be accomplished after hardening without grinding the root fillet radii. The full residual stress developed by case hardening is retained. The root surface finish will be as hobbed unless a grinding operation is incorporated.

A trochoidal fillet is produced by a protuberant hob or shaper cutter. (The undercut could be broached into the gear tooth. )

The protuberance cut gears are necessarily slightly thinner at the weakest section and have smaller root diameters as compared with full form ground gears; thus, the bending stress is increased. The material strength should also be greater. The resulting fatigue life, however, is not predictable because of the many factors involved which can not be accurately assessed.

A generated ground fillet was used for the full form gears to maintain similarity with the protuberant fillet configuration. All gears were shot peened in the root. The fillet type designation part of the designed experiment, therefore, included changes in tooth thickness, root diameter, case depth, and surface treatment. Figure 6 shows two typical fatigue test gears.

## MANUFACTURE OF FATIGUE TEST GEARS

Fatigue test gear manufacturing was controlled to minimize variation within and between each of the 16 groups. Significant efforts were made to maintain constant metallurgical microstructure and surface treatment as well as geometry. Specific items of control were as follows.

- All material was from a single heat (Carpenter Steel Company heat number 61629). The material was forged from 6-inch round corner squares to 2. 875- and 5. 125-inch bar stock form. The raw material record is given in Table V.
- All heat treat operations were performed at the same time except carburizing (due to two different case depths required) and stress relief after grinding (due to time limits).
- Copper plating prior to hardening and stripping of copper plate after hardening were each accomplished simultaneously on all parts.
- Shot blasting and peening were accomplished simultaneously on all gears of each group.
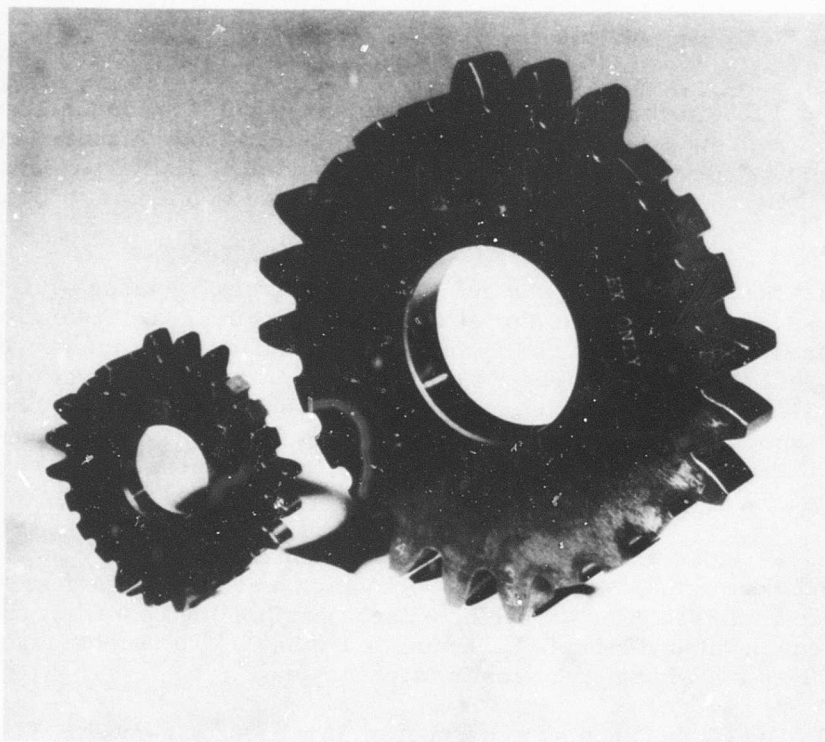
Figure 6. Typical Fatigue Test Gears.

- Gear tooth hobbing and grinding were accomplished by using an arbor that stacked all gears of each group. Each gear was honed separately.
- All test gears were black-oxide coated simultaneously (except for several sets which were processed early to permit initiation of testing).
- The high point of concentricity of all gears in each set was matched at each gear grinding operation, and gears were carefully aligned to obtain uniformity of stock removal.

TABLE V
RAW MATERIAL RECORD

Allison Purchase Order Numbers J8-05266 and J8-05265

STEEL SUPPLIER DATA—CARPENTER STEEL COMPANY

Material specification—AMS-6265
Heat number—61629
Material size—6-inch round corner squares
Grain size—5
Jominy hardenability—Top of ingot     $R_C 38$ at surface
                                                $R_C 38$ at 6/16 inch
                    Bottom of ingot     $R_C 39$ at surface
                                                  $R_C 38$ at 6/16 inch

TABLE V (CONT)
RAW MATERIAL RECORD

Hardness—Brinell 269
Jernkontoret (J. K.) rating

| Inclusion Type | A | | B | | C | | D | |
|---|---|---|---|---|---|---|---|---|
| Inclusion Size | Thin | Thick | Thin | Thick | Thin | Thick | Thin | Thick |
| Top | 1 | 0 | 1 | 0 | 0 | 0 | 1 | 0 |
| Bottom | 1 | 0 | 1 | 0 | 0 | 0 | 1 | 1 |

Chemical analysis

| C | Mn | P | S | Si | Cr | Ni | Mo |
|---|---|---|---|---|---|---|---|
| 0.11 | 0.66 | 0.004 | 0.004 | 0.30 | 1.33 | 3.39 | 0.14 |

Steel forger—Indianapolis Drop Forging Company Incorporated
Forged size—Two pieces 5.125 inches in diameter and 36 inches long
Two pieces 2.875 inches in diameter and 36 inches long

### ALLISON METALLURGICAL INSPECTION RECORD

Coarse etch—okay
Magnaflux step-down bars—okay
Chemical analysis

| C | Mn | P | S | Si | Cr | Ni | Mo |
|---|---|---|---|---|---|---|---|
| 0.10 | 0.67 | — | — | 0.29 | 1.29 | 3.41 | 0.12 |

Tensile tests

Material from 2.875-inch-diameter bar stock heat treated to Allison specification
(EPS 200) as follows: 1475°F. for 1 hour, oil quenched; 325°F. for 1 hour, air cooled;
Rockwell "C" hardness of 38.0 to 38.5. Tests were conducted at room temperature.

| Specimen number | Yield strength 0.2% offset (p.s.i.) | Tensile strength (p.s.i.) | Elongation in 1 inch (percent) | Reduction of area (percent) |
|---|---|---|---|---|
| A | 140,200 | 181,100 | 18.2 | 70.2 |
| B | 141,500 | 180,300 | 18.2 | 68.8 |
| C | 142,600 | 179,000 | 18.0 | 68.0 |

Izod impact tests

The heat treated material tests were conducted at room temperature.

| Specimen number | Impact energy (foot-pounds) | Reference |
|---|---|---|
| D | 74.0 | Russel, J. E., and Chesters, W.T., |
| E | 75.0 | "Significance of the Izod Test |
| F | 74.0 | with Regard to Gear Design and |
| | | Performance," Engineering, |
| | | Volume 176, 1953, pp. 166-169. |

Many in-process and finished part measurements were made to h.. define stock removal and to record the final geometry of each part. Tables VI and VII list the protuberant cut gear measurements and analysis. Tables VIII and IX provide comparable data for the ground fillet gears.

The root diameter, dimension over pins, root radius, and protuberance undercut depth are the critical dimensions for the fatigue specimens.

Most of the gears had some, usually slight, dimensional deviation. All the gears of each group were well within the dimensional tolerance limits. Thus, repeatabilily of fatigue test data within any group should be excellent due to the stack machining techniques employed. Some variation from the designed experiment, however, may occur between groups. These variations could be eliminated by basing bending stress calculations on actual rather than print dimensions.

Sample routing sheets for a full ground (EX-78772) and a protuberant cut gear (EX-78776) are given in Appendix II.

Table X lists the fatigue test gear hob dimensions necessary to define the gear tooth root fillet shape. The dimensions given must be modified by the finish stock allowance to obtain an accurate finished gear configuration. The full ground root fillet configuration hobs are listed to permit analysis of the finish stock allowance in the root fillet area rather than for bending stress determination.

## TEST RIG DESIGN AND PROCEDURE

The test rig was designed for single tooth fatigue testing of either the 2- or 4-inch-pitch-diameter gear. Single tooth testing was selected over a dynamic four-square gear tes' to permit accurate control of test variables. Adjacent teeth on the test gear were removed to ensure single tooth contact.

Two design concepts were considered for the fatigue testing device—a hydraulic servo-valve system where a measured torque is applied on the test gear to produce the desired tooth load and an electromagnetic shaker for use as the input loading device. The two concepts were evaluated on the basis of available equipment, usage experience, and inherent advantages and disadvantages. Design studies showed that the electromagnetic shaker was preferred, provided that a high frequency of operation could be achieved at the specified test loads. Additional considerations were accurate tooth load measurements and good dynamic stability.

To achieve the desired operational requirements, a fatigue test rig was designed with inherent high axial and radial stiffness of all load transmitting and reacting components and with a load cell at the point of tooth loading. The fatigue rig was coupled to an electromagnetic shaker. Operation at or near a system resonance of approximately 200 c.p.s. was realized. The principle of operation of the fatigue test rig is shown schematically in Figure 7.

The shaker driving force was applied directly to a mass which, in turn, loaded the gear tooth through a load cell. The mass was supported flexibly in the direction of loading and was stabilized in all radial directions by two disk-type flexible plates.

**TABLE VI**
**TABULATION OF PROTUBERANT FILLET GEAR MEASUREMENTS***

| Part Number | Root Fillet Radius | | | Root Diameter | | | | |
|---|---|---|---|---|---|---|---|---|
| | Print Minimum | After Hob | After Solution Machining | Print (± 0.002) | After Hob | After Solution Machining | Print | |
| EX-78776 | 0.050 | 0.060 to 0.065 | 0.065 to 0.070 | 3.533 | 3.535 | 3.5227 to 3.5241 | 4.3953 to 4.3999 | 4. |
| EX-78777 | 0.025 | 0.030 | 0.030 to 0.032 | 1.767 | 1.775 | 1.7679 to 1.7688 | 2.1953 to 2.2000 | 2. |
| EX-78778 | 0.080 | 0.085 | 0.090 | 3.533 | 3.536 | 3.5248 to 3.5275 | 4.3953 to 4.3999 | 4. |
| EX-78779 | 0.040 | 0.042 | 0.044 | 1.767 | 1.7745 | 1.7672 to 1.7682 | 2.1953 to 2.2000 | 2. |
| EX-78784 | 0.050 | 0.056 | 0.065 | 3.550 | 3.551 | 3.5412 to 3.5424 | 4.3973 to 4.4012 | 4. |
| EX-78785 | 0.025 | 0.026 to 0.032 | 0.028 to 0.036 | 1.775 | 1.7815 | 1.7755 to 1.7764 | 2.1967 to 2.2006 | 2. |
| EX-78786 | 0.067 | 0.068 to 0.070 | 0.070 to 0.075 | 3.550 | 3.555 | 3.5436 to 3.5448 | 4.3973 to 4.4012 | 4 |
| EX-78787 | 0.033 | 0.032 | 0.034 to 0.036 | 1.775 | 1.784 | 1.7775 to 1.7778 | 2.1967 to 2.2006 | 2 |

\* All dimensions in inches

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| Dimension Over Pins | | | | | |
|---|---|---|---|---|---|
| Print | After Hob | After Heat Treat | After Solution Machining | After Final Grind | Minimum Finishing Stock After Hob Operation |
| 4.3953 to 4.3999 | 4.4353 | 4.4338 to 4.4345 | 4.4201 to 4.4239 | 4.3963 to 4.3965 | 0.0354 |
| 2.1953 to 2.2000 | 2.2362 | 2.2300 to 2.2305 | 2.2246 to 2.2257 | 2.1958 to 2.1968 | 0.0362 |
| 4.3953 to 4.3999 | 4.4352 | 4.4339 to 4.4344 | 4.4205 to 4.4255 | 4.3903 to 4.3906 | 0.0353 |
| 2.1953 to 2.2000 | 2.2355 | 2.2347 to 2.2353 | 2.2247 to 2.2257 | 2.1961 to 2.1963 | 0.0355 |
| 4.3973 to 4.4012 | 4.431 | 4.4290 to 4.4298 | 4.4183 to 4.4205 | 4.3973 to 4.3980 | 0.0298 |
| 2.1967 to 2.2006 | 2.2306 | 2.2296 to 2.2305 | 2.2208 to 2.2222 | 2.1972 to 2.1978 | 0.0300 |
| 4.3973 to 4.4012 | 4.4316 | 4.4298 to 4.4300 | 4.4183 to 4.4202 | 4.3982 to 4.3983 | 0.0304 |
| 2.1967 to 2.2006 | 2.2312 | 2.2302 to 2.3209 | 2.2222 to 2.2230 | 2.1945 to 2.1949 | 0.0306 |

b

TABLE VII
## ANALYSIS OF PROTUBERANT FILLET GEAR MEASUREMENTS*

| | Root Diameter | | | | | | | Dimension Over P |
|---|---|---|---|---|---|---|---|---|
| Part Number | Maximum Change, Hob to Solution Machining | Maximum Variation Between Gears After Solution Machining | Finishing Stock After Hob Operation | Maximum Change, Hob to Heat Treat | Maximum Variation Between Gears After Heat Treat | Change Between Minimum Heat Treat and Minimum Solution Machining | Maximum Variation Between Gears After Solution Machining | Maxi. Cha. Hob Solut. Mach... |
| EX-78776 | 0.0123 | 0.0014 | 0.002 | 0.0015 | 0.0007 | 0.0137 | 0.0038 | 0.0 : |
| EX-78777 | 0.0071 | 0.0008 | 0.008 | 0.0062*** | 0.0005 | 0.0054 | 0.0011 | 0.0 1 |
| EX-78778 | 0.0118 | 0.0027 | 0.003 | 0.0013 | 0.0005 | 0.0134 | 0.0050 | 0.0 9 |
| EX-78779 | 0.0073 | 0.0010 | 0.0075 | 0.0008 | 0.0006 | 0.0100 | 0.0010 | 0.0 ( |
| EX-73784 | 0.0098 | 0.0012 | 0.001 | 0.0020 | 0.0008 | 0.0107 | 0.0022 | 0.0 : |
| EX-78785 | 0.0060 | 0.0009 | 0.0065 | 0.0010 | 0.0009 | 0.0088 | 0.0014 | 0.0 |
| EX-78786 | 0.0119 | 0.0012 | 0.005 | 0.0018 | 0.0002 | 0.0115 | 0.0019 | 0.0 |
| EX-78787 | 0.0065 | 0.0003 | 0.009 | 0.0010 | 0.0007 | 0.0080 | 0.0008 | 0.0 |
| Average † | 0.0115 | 0.0016 | 0.0024 | 0.0017 | 0.0006 | 0.0123 | 0.0032 | 0. |
| Average ‡ | 0.0067 | 0.0008 | 0.0078 | 0.0010 | 0.0007 | 0.0081 | 0.0011 | 0. |

* All dimensions in inches.

** Dimension over pins calculated for 0.000 to 0.004 backlash with mating gear on standard centers. Therefore, dimension over pins tolerances equivalent to 0.002 change in tooth thickness or 0.001 stock allowance per surf. The 0.0039 tolerance for 25-degree pressure angle gears and 0.0300 average finishing stock after hob are equ to 0.0077 per surface. The 0.0047 tolerance for 20-degree pressure angle gears and 0.0355 finishing stock af hob are equivalent to 0.0076 per surface.

*** Questionable reading—deleted from averages.

| | | | Dimension Over Pins | | | |
|---|---|---|---|---|---|---|
| Change Between Maximum Heat Treat and Minimum Solution Machining | Maximum Variation Between Gears After Solution Machining | Maximum Change Hob to Solution Machining | Change Between Minimum Solution Machining and Final Grind | Minimum Finishing Stock After Hob Operation ** | Maximum Variation Between Gears After Final Grind | Maximum Change, Hob to Final Grind |
| 0.0137 | 0.0038 | 0.0152 | 0.0238 | 0.0354 | 0.0002 | 0.0390 |
| 0.0054 | 0.0011 | 0.0116 | 0.0288 | 0.0362 | 0.0010 | 0.0404 |
| 0.0134 | 0.0050 | 0.0147 | 0.0302 | 0.0353 | 0.0003 | 0.0449 |
| 0.0100 | 0.0010 | 0.0108 | 0.0286 | 0.0355 | 0.00)2 | 0.0394 |
| 0.0107 | 0.0022 | 0.0127 | 0.0210 | 0.0298 | 0.0007 | 0.0337 |
| 0.0088 | 0.0014 | 0.0098 | 0.0236 | 0.0300 | 0.0006 | 0.0334 |
| 0.0115 | 0.0019 | 0.0133 | 0.0201 | 0.0304 | 0.0001 | 0.0334 |
| 0.0080 | 0.0008 | 0.0090 | 0.0277 | 0.0306 | 0.0004 | 0.0367 |
| 0.0123 | 0.0032 | 0.0140 | 0.0265 | — | 0.0003 | 0.0378 |
| 0.0081 | 0.0011 | 0.0103 | 0.0272 | -- | 0.0006 | 0.0375 |

† For large-diameter gears.

‡ For small-diameter gears.

ar on standard centers. Therefore,
s or 0.001 stock allowance per surface.
ge finishing stock after hob are equivalent
gears and 0.0355 finishing stock after

β

TABLE VIII
TABULATION OF GROUND FILLET GEAR MEASUREMENTS*

| Part Number | Root Fillet Radius | | | Root Diameter | | | Prin |
|---|---|---|---|---|---|---|---|
| | Print Minimum | After Hob | After Final Grind | Print (± 0.002) | After Hob | After Final Grind | Pri |
| EX-78772 | 0.050 | 0.075 | 0.065 | 3.5830 | 3.5916 | 3.5800 to 3.5806 (3.5830)** | 4.399 4.39 |
| EX-78773 | 0.025 | 0.040 | 0.040 | 1.7920 | 1.808 | 1.7836 to 1.7850 (1.7903)** | 2.195 2.20 |
| EX-78774 | 0.080 | 0.085 | 0.070 | 3.5830 | 3.594 | 3.5863 to 3.5882 (3.5820)** | 4.399 4.39 |
| EX-78775 | 0.040 | 0.036 to 0.038 | 0.034 | 1.7920 | 1.809 | 1.7950 to 1.7955 | 2.195 2.20 |
| EX-78780 | 0.050 | 0.065 to 0.070 | 0.055 to 0.060 | 3.600 | 3.6152 | 3.5998 to 3.6010 | 4.397 4.40 |
| EX-78781 | 0.025 | 0.026 | 0.026 to 0.028 | 1.800 | 1.815 | 1.8093 to 1.8105 | 2.196 2.20 |
| EX-78782 | 0.067 | 0.070 | 0.070 | 3.600 | 3.614 | 3.600 to 3.604 (3.605)** | 4.397 4.40 |
| EX-78783 | 0.033 | 0.032 to 0.036 | 0.034 to 0.036 | 1.800 | 1.815 | 1.805 (1.803)** | 2.196 2.20 |

*   All dimensions in inches.

**  Setup part not included.

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| | Root Diameter | | Dimension Over Pins | | | |
|---|---|---|---|---|---|---|
| rint | After Hob | After Final Grind | Print | After Hob | After Heat Treat | After Finish Grind and Hone |
| 999 to 3953 | 3.5916 | 3.5800 to 3.5806 (3.5830)** | 4.3999 to 4.3953 | 4.4354 | 4.4345 to 4.4350 | 4.3961 to 4.3971 (4.396)** |
| 953 to 2000 | 1.808 | 1.7836 to 1.7850 (1.7903)** | 2.1953 to 2.2000 | 2.2344 | 2.2335 to 2.2342 | 2.1920 to 2.1922 (2.1942)** |
| 999 to 3953 | 3.594 | 3.5863 to 3.5882 (3.5820)** | 4.3999 to 4.3953 | 4.4352 to 4.4354 | 4.4340 to 4.4347 | 4.3990 to 4.3990 (4.3941)** |
| 953 to 2000 | 1.809 | 1.7950 to 1.7955 | 2.1953 to 2.2000 | 2.2355 | 2.2345 to 2.2355 | 2.1912 to 2.1928 (2.1895)** |
| 973 to 4012 | 3.6152 | 3.5998 to 3.6010 | 4.3973 to 4.4012 | 4.4293 to 4.4298 | 4.4275 to 4.4282 | 4.3997 to 4.4005 |
| 967 to 2006 | 1.815 | 1.8093 to 1.8105 | 2.1967 to 2.2006 | 2.2312 to 2.2313 | 2.2305 to 2.2307 | 2.1961 to 2.1976 |
| 973 to 4012 | 3.614 | 3.600 to 3.604 (3.605)** | 4.3973 to 4.4012 | 4.4319 | 4.4292 to 4.4297 | 4.3976 to 4.3981 (4.3967)** |
| 967 to 2006 | 1.815 | 1.805 (1.803)** | 2.1967 to 2.2006 | 2.2305 | 2.2295 to 2.2300 | 2.1965 to 2.1972 (2.1947)** |

B

## TABLE IX
## ANALYSIS OF GROUND FILLET GEAR MEASUREMENTS*

| Part Number | Root Diameter | | | Maximum Change, Hob to Heat Treat | Maximum Variation Between Gears After Heat Treat | Maximum Change, Minimum Heat Treat to Minimum Hone |
|---|---|---|---|---|---|---|
| | Maximum Change, Hob to Final Grind | Maximum Variation Between Gears after Final Grind | Grind Stock After Hob Operation (±0.002) | | | |
| EX-78772 | 0.0116 | 0.0006 | 0.0086 | 0.0009 | 0.0005 | 0.0384 |
| EX-78773 | 0.012 | 0.000 | 0.016 | 0.0009 | 0.0007 | 0.0415 |
| EX-78774 | 0.0077 | 0.0019 | 0.011 | 0.0012 | 0.0007 | 0.0370 |
| EX-78775 | 0.014 | 0.0005 | 0.017 | 0.0010 | 0.0010 | 0.0433 |
| EX-78780 | 0.0154 | 0.0012 | 0.0152 | 0.0018 | 0.0007 | 0.0278 |
| EX-78781 | 0.0057 | 0.0012 | 0.015 | 0.0007 | 0.0002 | 0.0344 |
| EX-78782 | 0.014 | 0.0040 | 0.014 | 0.0027 | 0.0005 | 0.0316 |
| EX-78783 | 0.010 | 0.000 | 0.015 | 0.0010 | 0.0005 | 0.033 |
| † Average | 0.0122 | 0.0019 | 0.0122 | 0.0017 | 0.0006 | 0.0337 |
| ‡ Average | 0.0104 | 0.0009 | 0.016 | 0.0009 | 0.0006 | 0.0381 |

* All dimensions in inches.

** Dimension over pins calculated for 0.000 to 0.004 backlash with mating gear on standard centers. pins tolerances equivalent to 0.002 change in tooth thickness or 0.001 stock allowance per surface. 25-degree pressure angle gears and 0.0300 average finishing stock after hob are equivalent to 0.00 tolerance for 20-degree pressure angle gears and 0.0355 finishing stock after hob are equivalent to

† For large-diameter gears.

‡ For small-diameter gears.

| D... | Max Variation Between Gears After Grind a... | ...mum ...ge, ... to After ...Treat | Maximum Variation Between Gears After Heat Treat | Maximum Change, Minimum Heat Treat to Minimum Hone | Maximum Variation Between Gears After Final Grind and Hone | Maximum Change, Hob to Final Grind and Hone | Maximum Finishing Stock After Hob Operation ** | Pressure Angle (Degrees) |
|---|---|---|---|---|---|---|---|---|
| | 0. | 09 | 0.0005 | 0.0384 | 0.0010 | 0.0393 | 0.0355 | 20 |
| | 0. | 09 | 0.0007 | 0.0415 | 0.0002 | 0.0424 | 0.0344 | 20 |
| | 0. | 12 | 0.0007 | 0.0370 | 0.0018 | 0.0384 | 0.0353 | 20 |
| | 0. | 10 | 0.0010 | 0.0433 | 0.0016 | 0.0443 | 0.0355 | 20 |
| | 0. | 18 | 0.0007 | 0.0278 | 0.0008 | 0.0296 | 0.0281 | 25 |
| | 0. | 07 | 0.0002 | 0.0344 | 0.0015 | 0.0351 | 0.0306 | 25 |
| | C. | 27 | 0.0005 | 0.0316 | 0.0005 | 0.0343 | 0.0307 | 25 |
| | 0. | 10 | 0.0005 | 0.033 | 0.0007 | 0.0340 | 0.0299 | 25 |
| | 0. | 17 | 0.0006 | 0.0337 | 0.0010 | 0.0354 | -- | — |
| | 0. | 09 | 0.0006 | 0.0381 | 0.0010 | 0.0389 | — | — |

Theref.　　with mating gear on standard centers.  Therefore, dimension over
e.  The C　　s or 0.001 stock allowance per surface.  The 0.0039 tolerance for
0077 per ...　ig stock after hob are equivalent to 0.0077 per surface.  The 0.0047
to 0.0070　　nishing stock after hob are equivalent to 0.0076 per surface.

β

## TABLE X
## HOB DIMENSIONS

| Gear Configuration | Gear Part Number | Hob Tooth Thickness HTT (inches) | Hob Addendum HADD (inches) | Hob Lead, HLEAD (inches) | Hob Pressure Angle, HPAR (degrees) | Hob Tip Radius, HTIPR (inch( |
|---|---|---|---|---|---|---|
| 1 | EX-78772 | 0.2468 | 0.2005 | 0.52436 | 20 | 0.055 to 0.050 |
| 2 | EX-78773 | 0.1159 | 0.0962 | 0.26194 | 20 | 0.025 to 0.030 |
| 3 | EX-78774 | 0.2468 | 0.2005 | 0.52436 | 20 | 0.072 full |
| 4 | EX-78775 | 0.1159 | 0.0962 | 0.26194 | 20 | 0.033 full |
| 5 | EX-78776 | 0.2032 | 0.1717 | 0.50888 | 14.5 | 0.050 to 0.055 |
| 6 | EX-78777 | 0.0943 | 0.0842 | 0.25421 | 14.5 | 0.025 |
| 7 | EX-78778 | 0.2032 | 0.1717 | 0.50888 | 14.5 | 0.082 full |
| 8 | EX-78779 | 0.0943 | 0.0842 | 0.25421 | 14.5 | 0.039 full |
| 9 | EX-78780 | 0.2468 | 0.1920 | 0.52435 | 25 | 0.045 to 0.040 |
| 10 | EX-78781 | 0.1159 | 0.0920 | 0.26194 | 25 | 0.024 full |
| 11 | EX-78782 | 0.2468 | 0.1920 | 0.52435 | 25 | 0.053 full |
| 12 | EX-78783 | 0.1159 | 0.0920 | 0.26194 | 25 | 0.024 full |
| 13 | EX-78784 | 0.1799 | 0.1509 | 0.50564 | 20 | 0.050 to 0.055 |
| 14 | EX-78785 | 0.0654 * | 0.0500 * | 0.24632 | 15.5 | 0.025 to 0.030 |
| 15 | EX-78786 | 0.1449 * | 0.1030 * | 0.49301 | 15.5 | 0.067 full |
| 16 | EX-78787 | 0.0654 * | 0.0500 * | 0.24632 | 15.5 | 0.032 full |

\* Theoretical

A

| Hob Pressure Angle, PAR (degrees) | Hob Tip Radius, HTIPR (inches) | Hob Protuberance, HPW (inches) | Hob Part Number | Tooth Thickness per Side (inches) | Root Diameter per Side (inches) |
|---|---|---|---|---|---|
| 20 | 0.055 to 0.050 | 0 | SPT-2603 | 0.008 | 0.008 |
| 20 | 0.025 to 0.030 | 0 | SPT-2608 | 0.008 | 0.008 |
| 20 | 0.072 full | 0 | SPT-2602 | 0.008 | 0.008 |
| 20 | 0.033 full | 0 | SPT-2607 | 0.008 | 0.008 |
| 14.5 | 0.050 to 0.055 | 0.007 to 0.008 | SPT-2604 | 0.008 | 0.003 |
| 14.5 | 0.025 | 0.0055 to 0.0060 | SPT-2611 | 0.008 | 0.003 |
| 14.5 | 0.082 full | 0.006 to 0.007 | SPT-2605 | 0.008 | 0.003 |
| 14.5 | 0.039 full | 0.0050 to 0.0055 | SPT-2609 | 0.008 | 0.003 |
| 25 | 0.045 to 0.040 | 0 | SPT-2594 | 0.008 | 0.008 |
| 25 | 0.024 full | 0 | SPT-2597 | 0.008 | 0.008 |
| 25 | 0.053 full | 0 | SPT-2595 | 0.008 | 0.008 |
| 25 | 0.024 full | 0 | SPT-2598 | 0.008 | 0.008 |
| 20 | 0.050 to 0.055 | 0.007 to 0.008 | SPT-2593 | 0.008 | 0.003 |
| 15.5 | 0.025 to 0.030 | 0.007 to 0.008 | SPT-2600 | 0.008 | 0.003 |
| 15.5 | 0.067 full | 0.007 to 0.008 | SPT-2591 | 0.008 | 0.003 |
| 15.5 | 0.032 full | 0.007 to 0.006 | SPT-2599 | 0.008 | 0.003 |

B
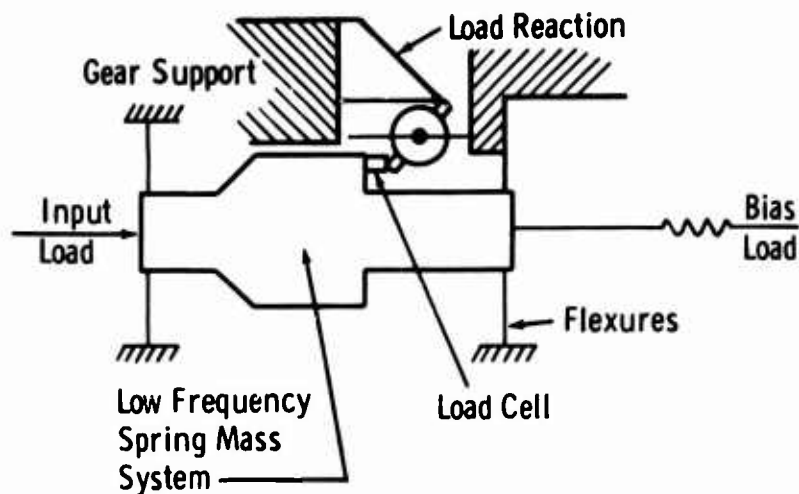
Figure 7. Principle of Operation of Fatigue Test Rig.

The required static preload was provided by compressing a relatively low spring rate coil spring. Inertia loading of the tooth, using the moving mass, made possible considerable force amplification at and near the system axial resonance. The forced dynamic load was about the mean value which, in this case, was the static preload. Figure 8 shows the test rig in its final configuration. Figure 9 shows the rig coupled to the shaker.

The load cell incorporated at the point of tooth loading to provide accurate control of both static and dynamic tooth loading during fatigue testing was an Allison designed strain gage type cell. Figure 10 shows the load cell instrumented with axial and circumferential strain gages, and Figure 11 shows the load cell in its final assembly. The strain gage hookup was a four-active-arm bridge. The bridge signal output was directly proportional to the change in applied thrust, independent of load cell bending and temperature change, and $2(1 + \mu)$ times as large as the corresponding output of a single strain gage. The symbol $\mu$ is Poisson's ratio.

The automatic control system of the electromagnetic shaker was not used. Excellent control stability was realized by manual control.

A series of check-out procedures was performed prior to dynamic testing. The following paragraphs present the check-out procedures in the sequence in which they were performed.

● Radial Spring Rate of Fatigue Rig

The fatigue rig was installed in the electromagnetic shaker and instrumented with dial indicators as shown in Figure 12. With gear EX-78784 installed and statically loaded by means of the bias spring loading device, the radial deflections were measured. The radial spring rate of the system as determined by test was 5,900,000 pounds/inch. This high radial spring rate verified the design objective of high system stiffness to ensure accurate load application at the high point of single tooth contact and good alignment of all moving parts during operation.

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● Dimensional Check-Out

**Measurements** were made to verify that contact between the load member tip and **the gear tooth occurred** at the high point of single tooth contact. The measurements **verified tip spacing** to the center of the pilot shaft to be as designed, and to ensure **tip contact** at the high point of single tooth contact during fatigue. Figures 13 and **14 show** typical dimensions for the 6- and 12-pitch gears.

● Tooth Load Distribution

Gear EX-78784 was designated as the check-out gear. The gear was instrumented with strain gages and a thermocouple, as shown in Figure 15. The instrumented gear was installed in the fatigue test rig, and a static load was applied in 1000-pound increments to 3000 pounds. The strain read-out of the two gages on face A was compared for indication of nonuniform loading or misalignment. The gages indicated uniform loading and good alignment. Accurate location of the strain gages was verified by inserting a small piece of shim stock, 0.003 inch thick, between the load member tip and the gear tooth. The shim stock was inserted an equal distance on both sides of the gear tooth, and differential strain was compared. The differential strain was of equal value, verifying good strain gage location.

● Dynamic Resonance Frequency

To determine the system operating frequency, a frequency scan was made versus shaker driver current. With the check-out gear installed and preloaded to 1000 pounds, the frequency scan was made from 50 to 500 c.p.s., plotting driver current while dynamically applying ±800 pounds of load to the gear tooth. The frequency scan indicated that the system resonance frequency was 240 c.p.s. with a reduction of 20:1 in driver coil current at resonance. Figure 16 shows the relative response.

● Dynamic Separation

To ensure continued contact between the gear tooth and the load member tip and to determine differential load margin, the output signal of a dynamic gage on face B was displayed on an oscilloscope. By varying the dynamic load about a constant preload, the signal wave shape was analyzed. Figure 17 presents the pictorial wave shape analysis. The analysis shows that a minimum of 20 pounds differential is required to maintain contact between the tooth and load tip.

● Load Cell Calibration

To eliminate inaccuracies in the loading, a precise calibration was made on the load cell. The load cell was tested in a Baldwin press as shown in Figures 18 and 19. The load was applied in 500-pound increments to 5000 pounds maximum; the output of the strain gage bridge was recorded. Each load cell was tested five times for repeatability. Figure 20 shows typical calibration data. The calibration of the load cell repeated within one percent in the new condition and within two percent after usage.

To allow the load member tip to contact the gear test tooth at the high point of single tooth contact, a number of teeth were removed as shown in Figure 21. Figure 21 shows load sides A and B. Teeth 1, 2, 3, and 4 are the test teeth, and teeth 1X, 2X, 3X, and 4X are the load reaction teeth.

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Load Member and Force Gage

A

4-Inch Test Gear

Section A-A

2-Inch Test Gear
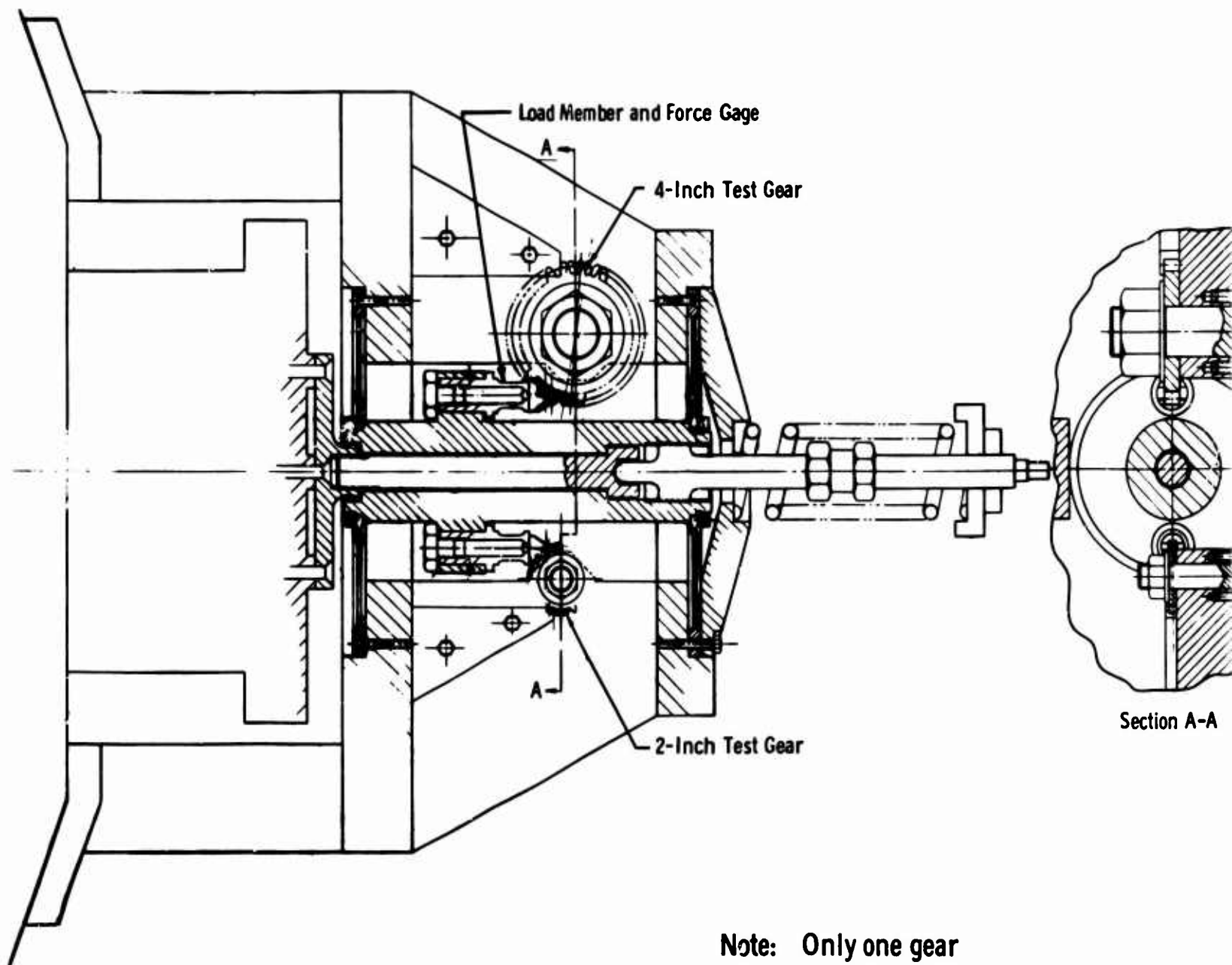
Note: Only one gear
tested at a time.

Figure 8. Fatigue Test Rig Schematic.

31

Section A-A

ir
me.

ß
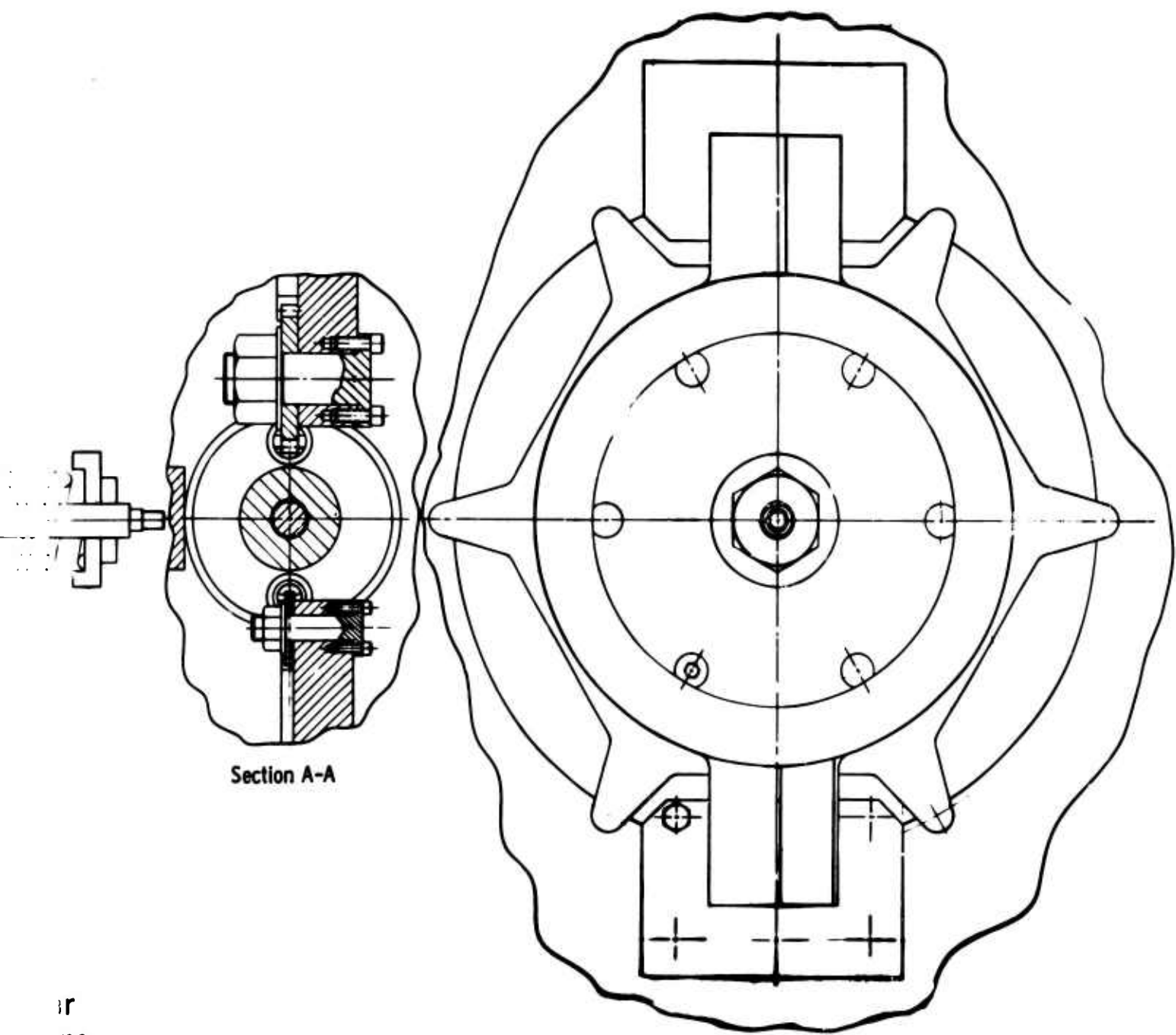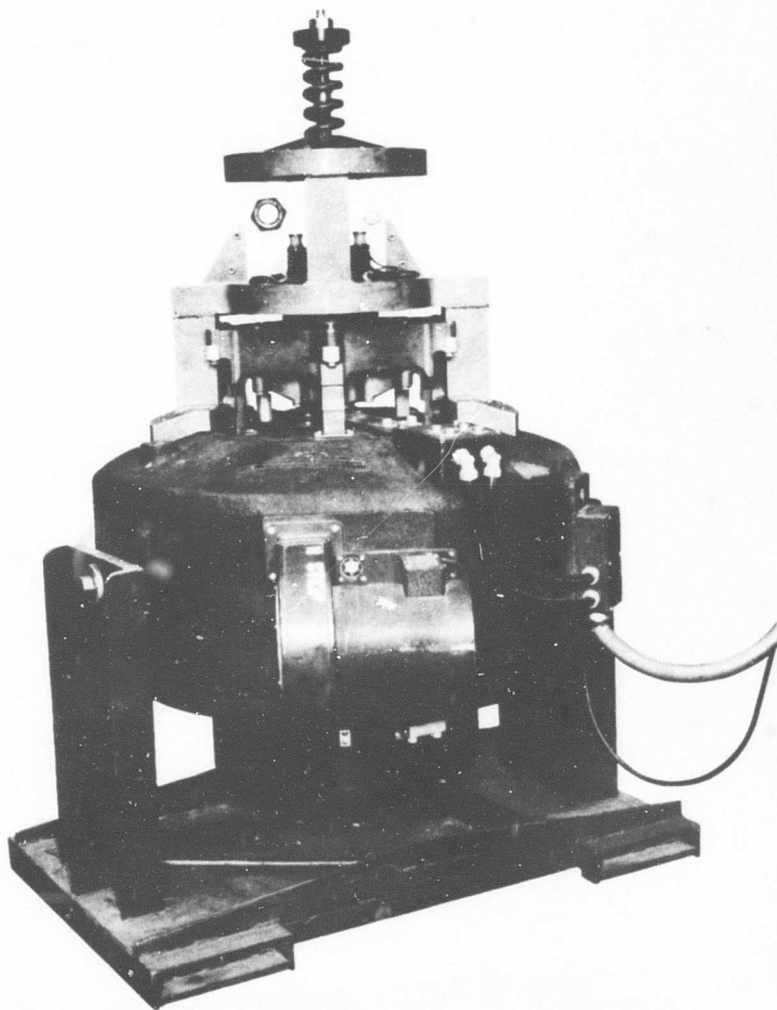
Figure 9.  Fatigue Test Setup.
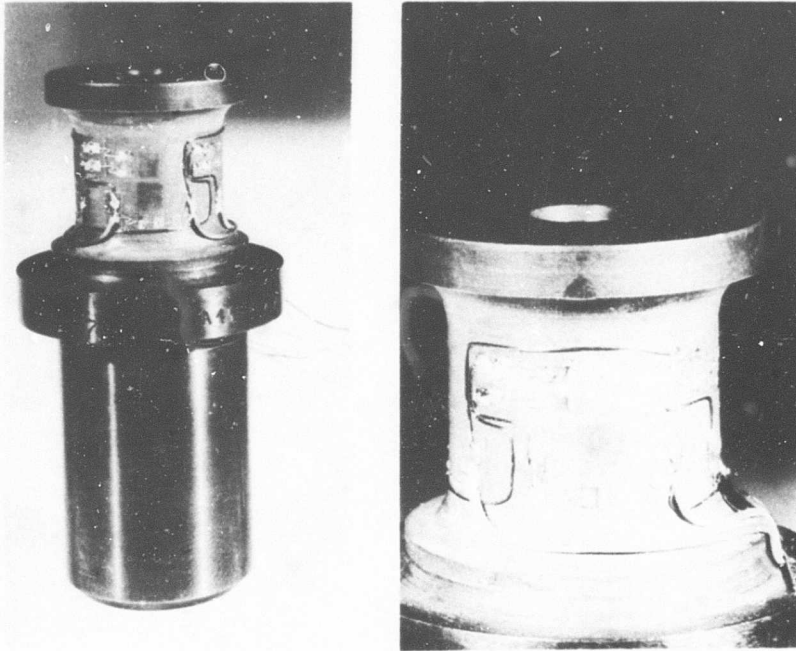
Figure 10. Load Cell Showing Instrumentation.
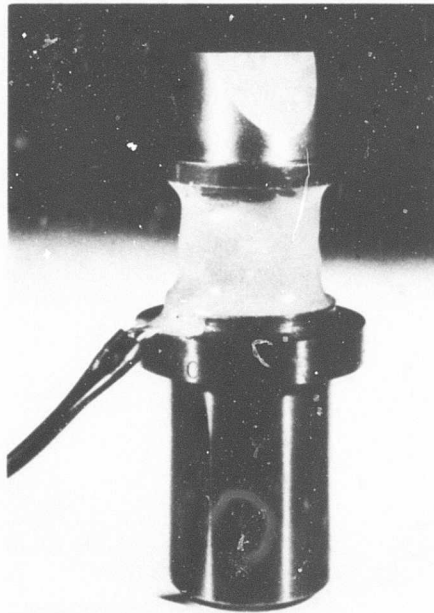


Figure 11. Assembled Load Cell.
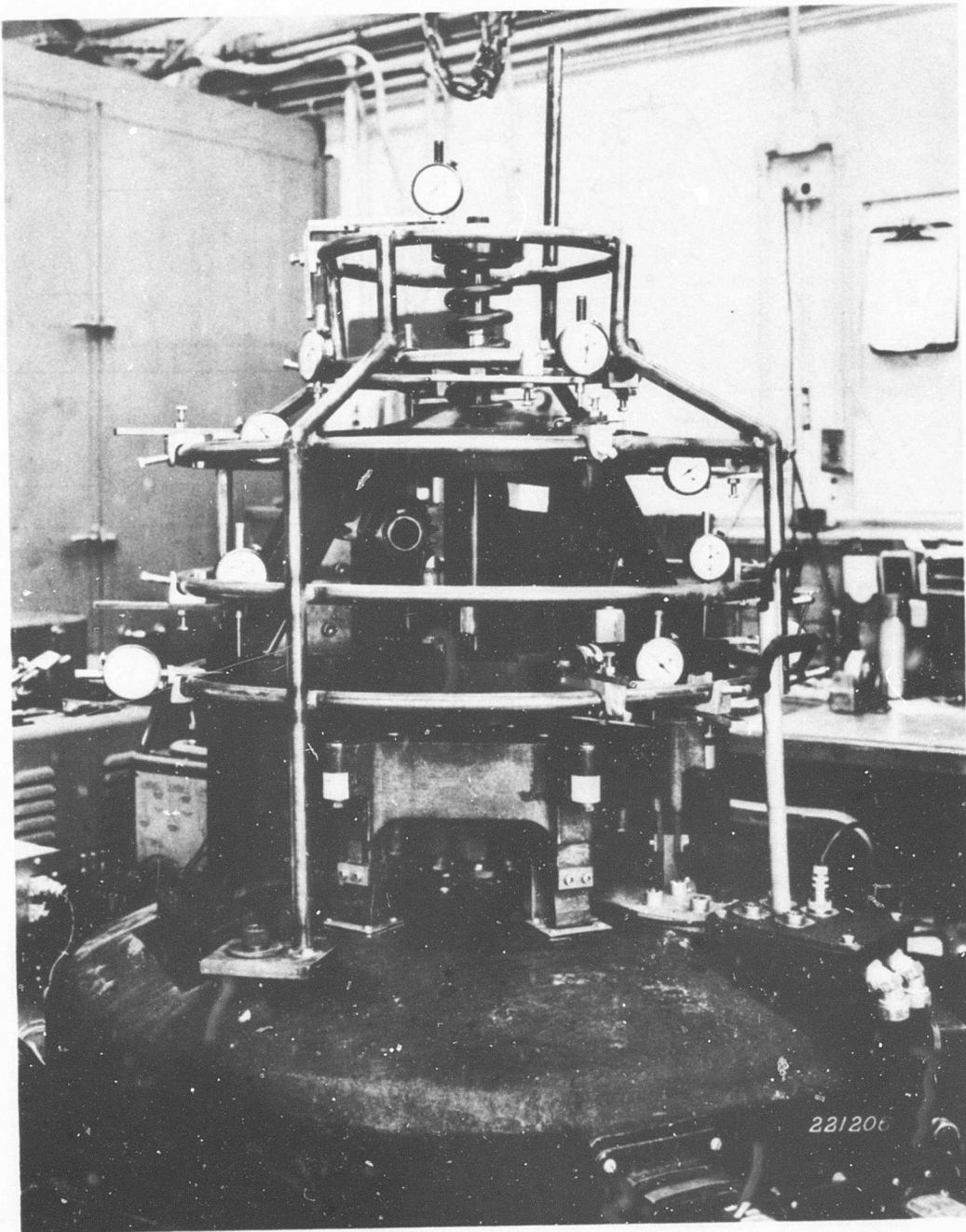
Figure 12. Instrumented Fatigue Test Rig.

**EX-78786**
**Pressure Angle—25 Degrees**
**Diametral Pitch—6**

Fatigue Test Gear

1.770R

0.9918

Load Cell

Load Reaction Block

1.813

Loading Arbor

2.3125
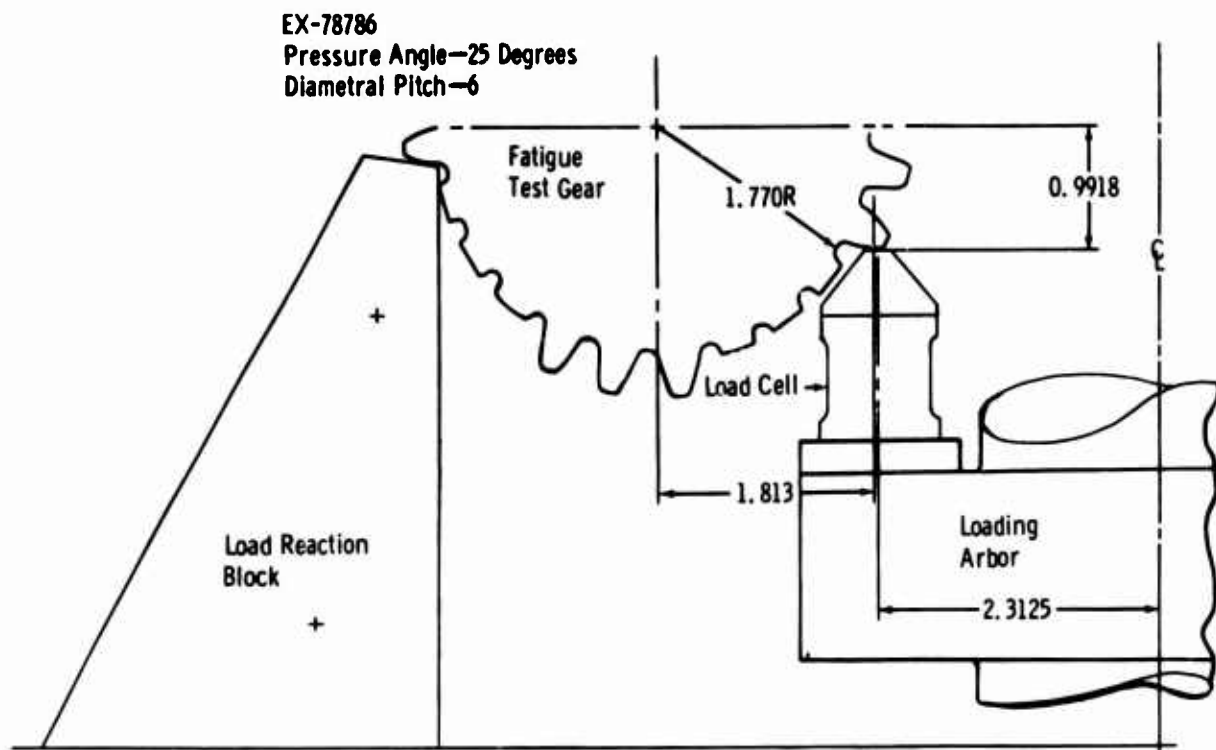
Figure 13.  Typical Dimensions of 6-Pitch Gear Test Setup.



**EX-78783**
**Pressure Angle—25 Degrees**
**Diametral Pitch—12**

0.5028

0.900 R

Load Cell

0.9063

Loading Arbor

2.3125

Load Reation Block
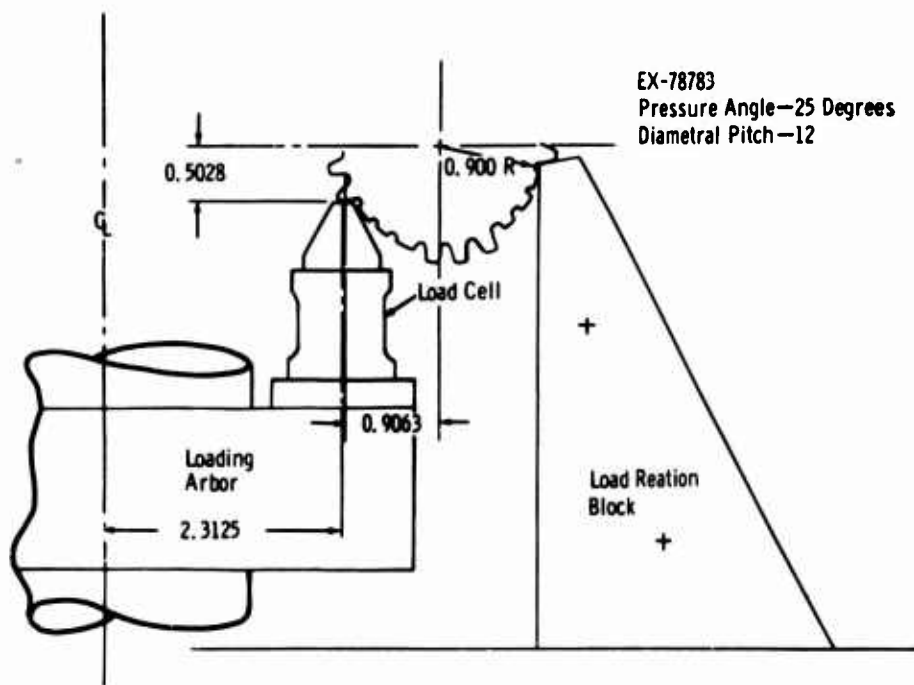
Figure 14.  Typical Dimensions of 12-Pitch Gear Test Setup.
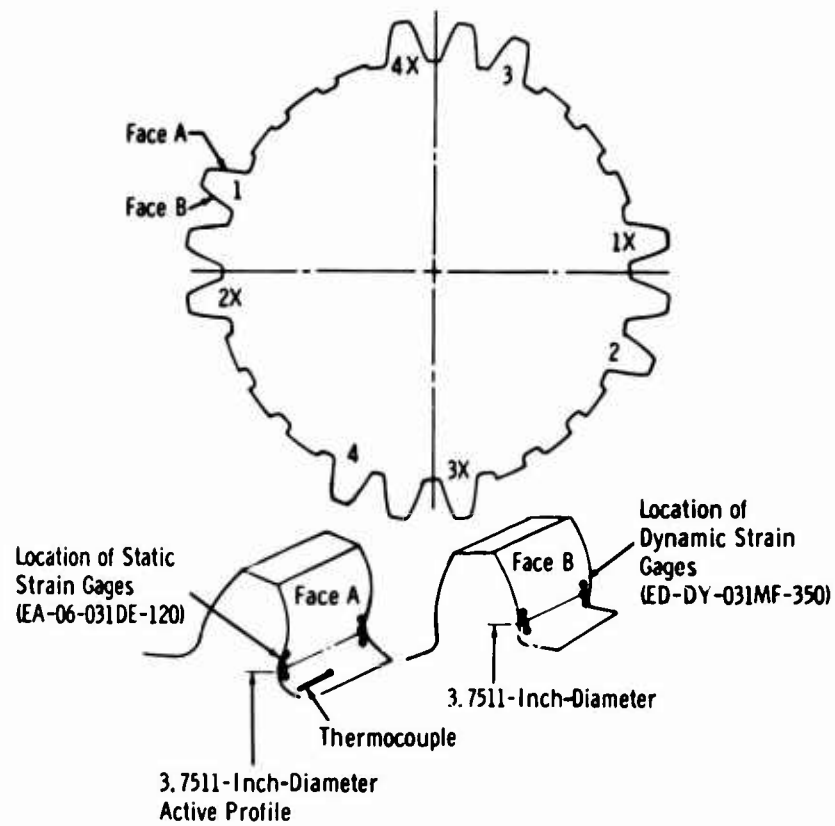
Figure 15. Schematic of Check-Out Gear Instrumentation.
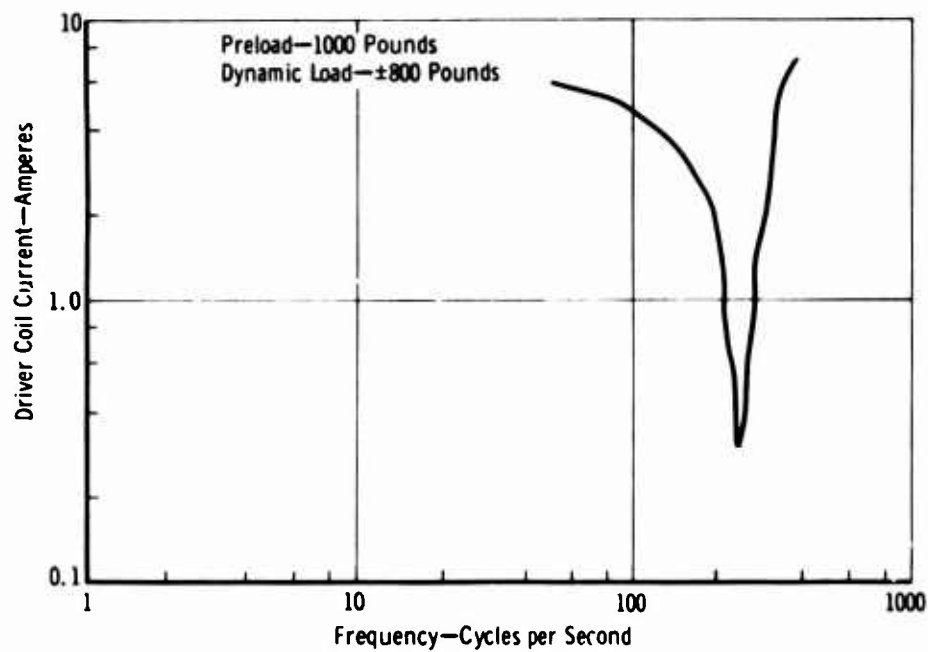


Figure 16. Test System Resonant Frequency.
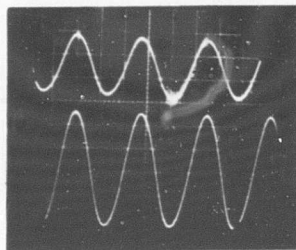
Load Member
Bridge

Dynamic
Strain Gage

Static Preload—1320 Pounds
Alternating Load—±1230 Pounds
No Separation
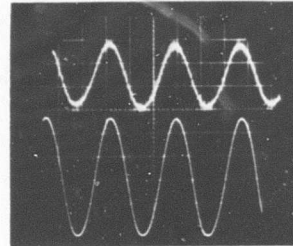
Static Preload—1320 Pounds
Alternating Load—±1310 Pounds
No Separation
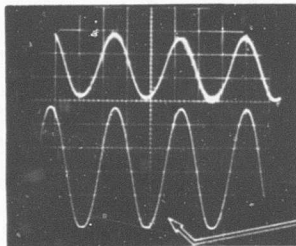
Flat
Peak

Static Preload—1320 Pounds
Alternating Load—±1345 Pounds
Separation
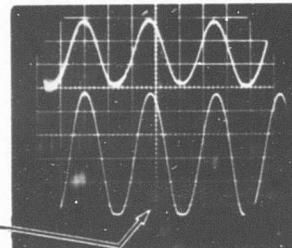
Static Preload—1320 Pounds
Alternating Load—±1380 Pounds
Separation

Figure 17.  Dynamic Strain Gage Signal Showing Tooth-to-Load Tip Contact.
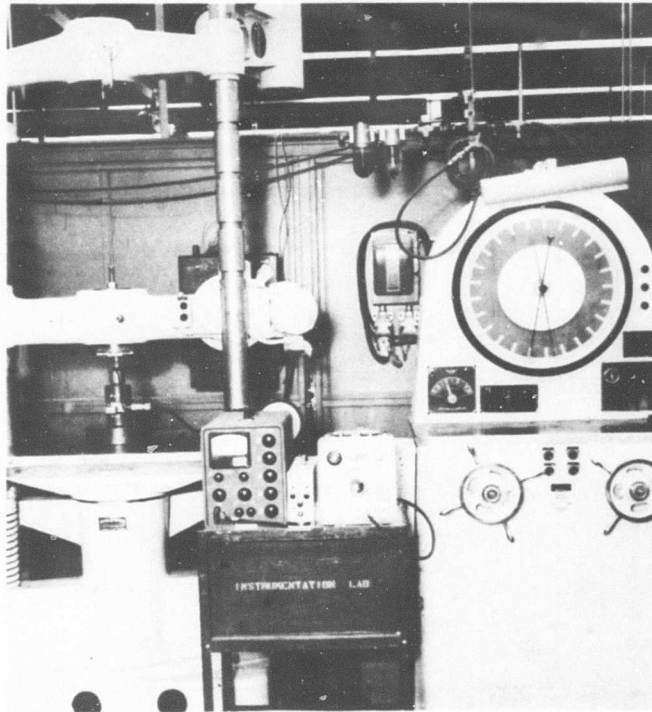
Figure 18.  Load Cell Test Setup.
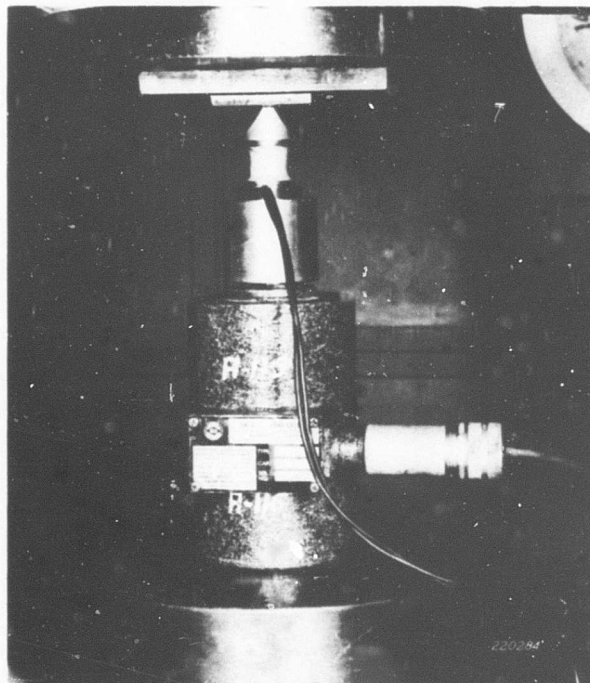


Figure 19.  Close-up of Load Cell Test Setup.
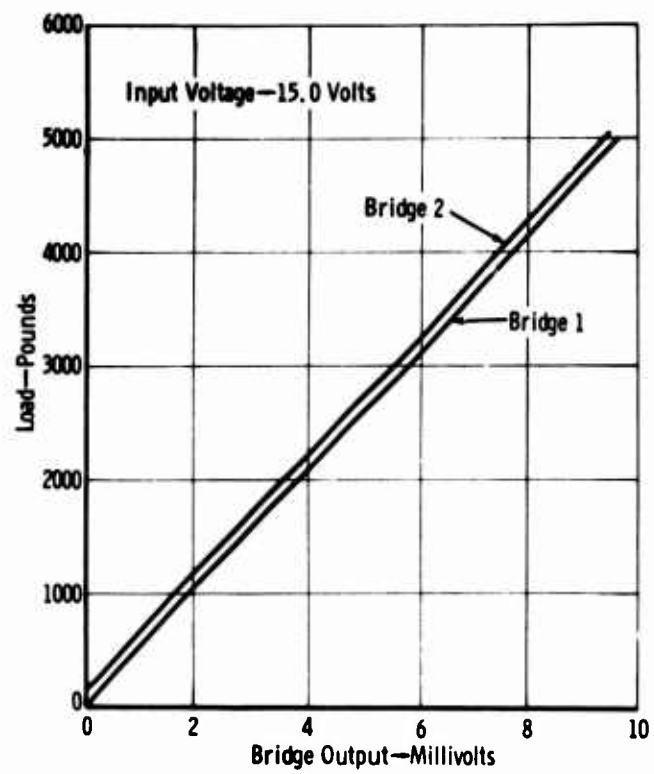
Figure 20. Typical Load Cell Calibration Curve.
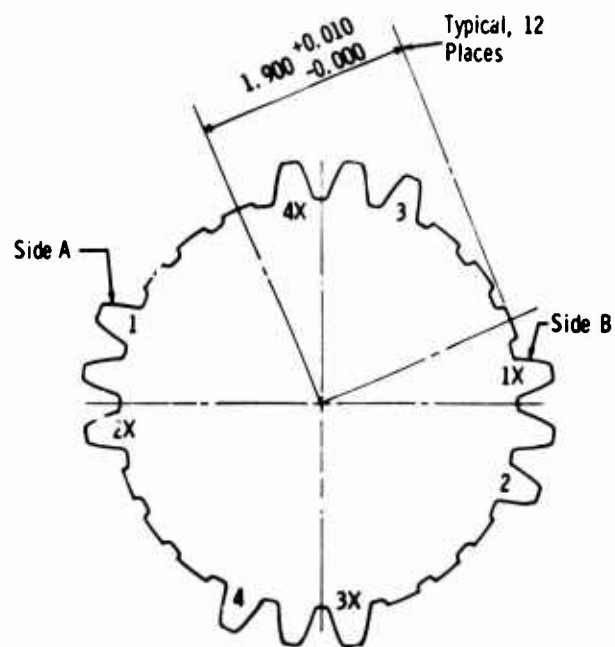


Figure 21. Test Gear Showing Teeth Removed.

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The test procedure required that the test tooth, once positioned, be preloaded with a bias load which was equal to one-half of the total fatigue load. Once the preload was obtained and verified by the load cell, an alternating load was applied about a mean which was the preload. The tentative plan was that three gear teeth be tested for each combination of variables until fatigue failure occurred or $10^7$ cycles were accumulated.

During testing, the dynamic load at the load cell (signal from strain gage bridge) was monitored and recorded on a strip chart recorder. A typical strip chart recording is shown in Figure 22.
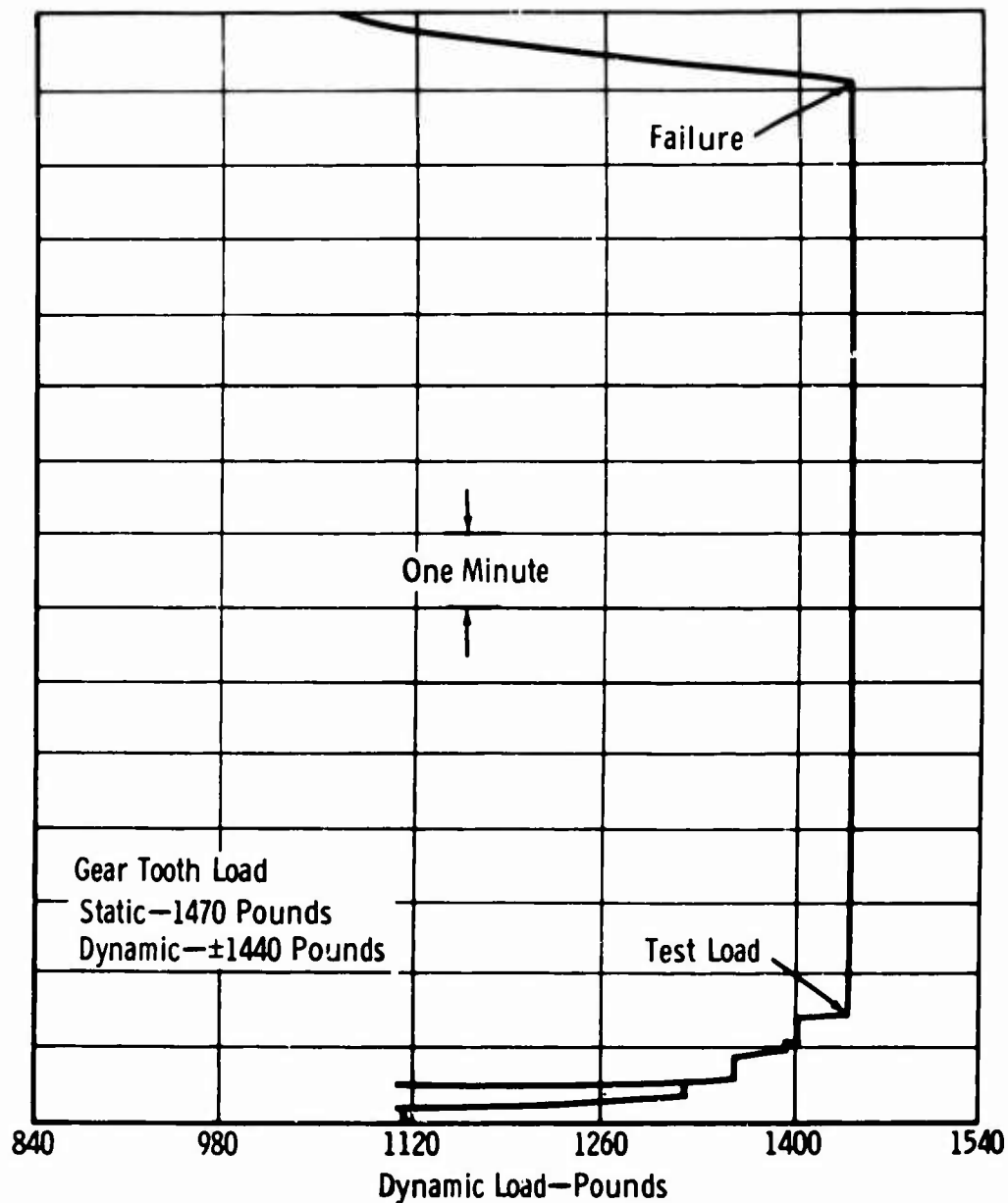


Figure 22. Typical Strip Chart Recording of Test Gear Dynamic Load.

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## RESULTS

### FATIGUE TESTS

The fatigue test program was based on a designed experiment for evaluation of four geometric variables—diametral pitch, pressure angle, root fillet size, and root fillet configuration. Two levels of each variable were employed requiring 16 different gear configurations. See Table IV. Initially, three teeth from each gear configuration were to be tested at four stress levels. Failures were required to permit test evaluation on the finite portion of the S/N curve. Early test experience with the small 12 diametral pitch gears indicated only a 30-percent spread between the desired minimum and maximum stress levels. The maximum stress was determined by the short test time (3 to 5 minutes) and high stresses that could cause plastic yielding and thus result in a different mode of failure. The minimum stress was determined by a high percent of runouts to 10,000,000 cycles without failure. It was decided, therefore, to obtain four failures at three stress levels to permit a 10-percent difference between levels.

Table XI lists the fatigue test data—load, cycles to failure, and configuration—for the 214 gear teeth tested. Of this total, 173 failed; the remaining gear tooth tests were terminated at $2 \times 10^6$ or $10^7$ cycles.

Fatigue test data for each configuration are plotted as S/N curves based on unit load in Figures 23 through 38. Unit load is defined as the equivalent load in pounds on a tooth having a diametral pitch of 1 and a face width of 1 inch. The mean curve drawn through the data was calculated by a procedure explained in detail in Appendix III. Proportionality factors can be used to relate applied load (test rig load), unit load, Lewis stress, Dolan-Broghamer stress, AGMA stress, Heywood stress, and Kelley-Pedersen stress for any single gear configuration. Therefore, S/N curves of the test data based on any of these stress calculation methods would produce the same fit of the mean curve to the data points. S/N curves based on AGMA calculated stress are presented in Appendix IV.

A series of reworks was initiated during the test program to modify or perfect parts related to the fatigue rig. The areas involved are discussed in the following paragraphs.

### Cooling Air

As a result of the high fatigue loads required for the gears having a diametral pitch of 6, it became necessary to provide cooling air to the fatigue tooth at the tension fillet and lubrication between the tooth and load cell tip. The need for cooling air at the compression fillet became apparent when two gears cracked from the tooth root to the gear center. Metallurgical analysis indicated that high localized temperatures existed during the final phase of tooth fatigue. Additional cooling air eliminated this problem. All but three teeth on the large gears were tested with the additional cooling air. It is believed that the test results for these three teeth were not seriously biased.

### Tip

The initial design specified that the contact surfaces of the tips be coated with plasma spray tungsten carbide. The process was to provide a surface which would offer resistance to wear, scuffing, and distortion. However, after limited usage, the coating cracked and cavitated. The first rework, nitriding the contact surface, was an improvement under low-load conditions, but the surface distorted under high loads. The second

43
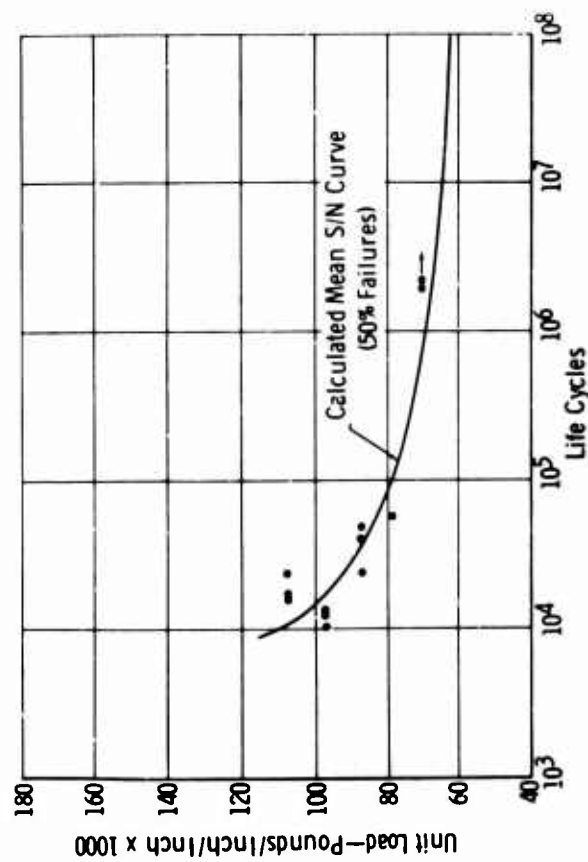
Figure 23. Fatigue Test Results—EX-78772.

Figure 24. Fatigue Test Results—EX-78773.

Figure 25. Fatigue Test Results—EX-78774.
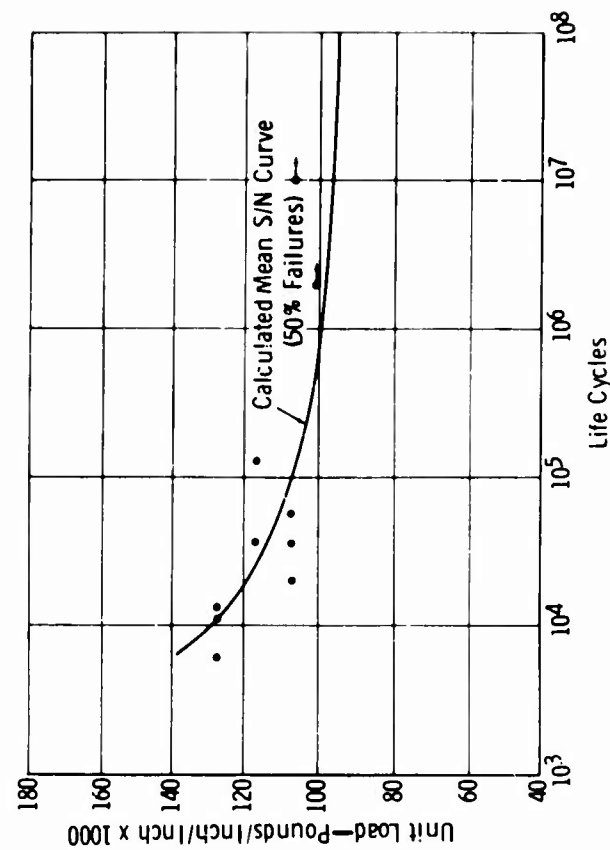
Figure 26. Fatigue Test Results—EX-78775.

44
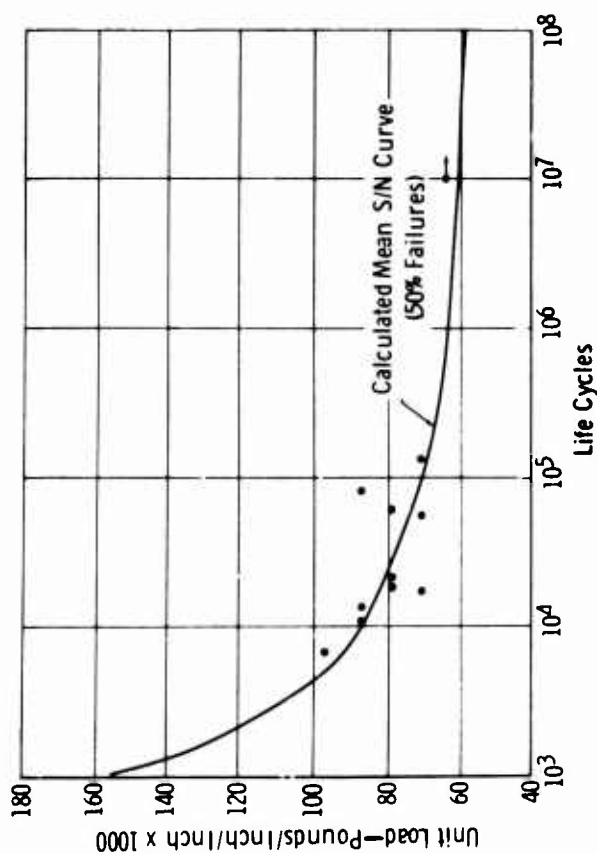
Figure 27. Fatigue Test Results—EX-78776.



Figure 28. Fatigue Test Results—EX-78777.



Figure 29. Fatigue Test Results—EX-78778.
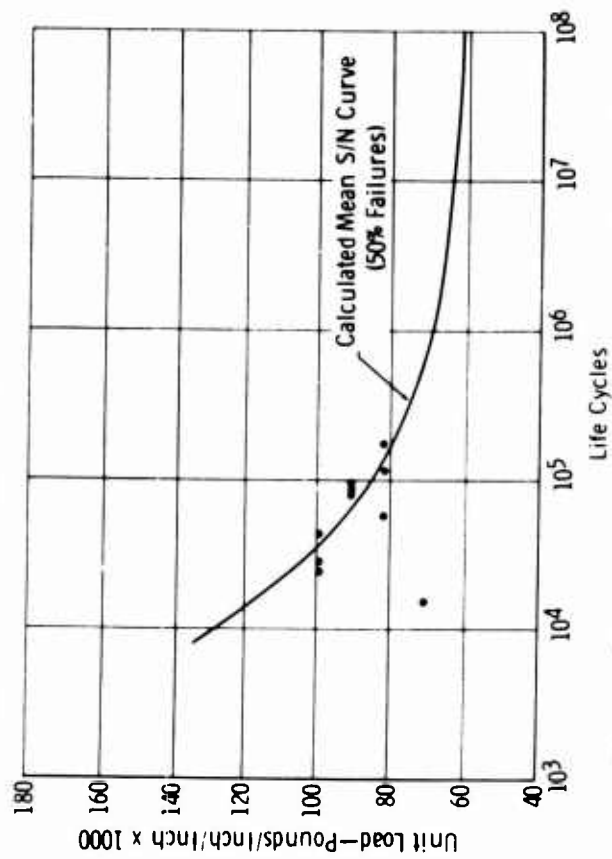


Figure 30. Fatigue Test Results—EX-78779.

45

Figure 32. Fatigue Test Results—EX-78781.
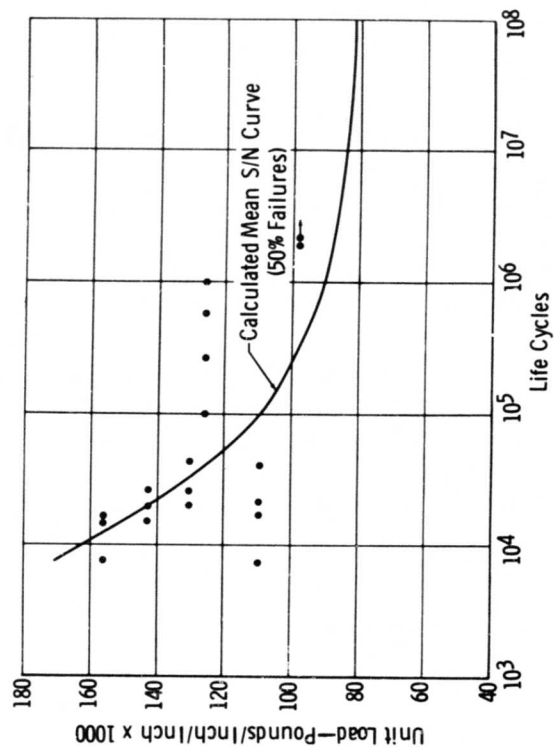


Figure 34. Fatigue Test Results—EX-78783.



Figure 31. Fatigue Test Results—EX-78780.



Figure 33. Fatigue Test Results—EX-78782.

46
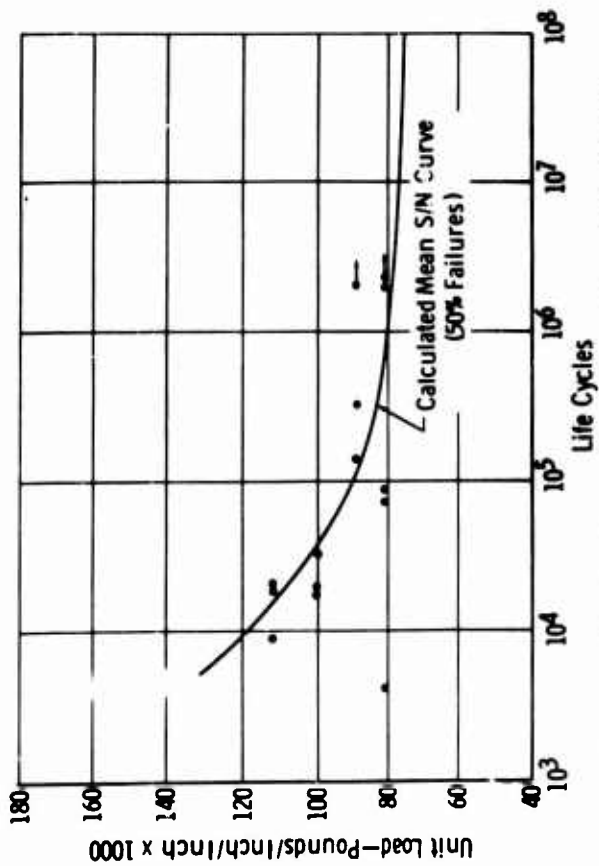
Figure 35. Fatigue Test Results—EX-78784.



Figure 36. Fatigue Test Results—EX-78785.
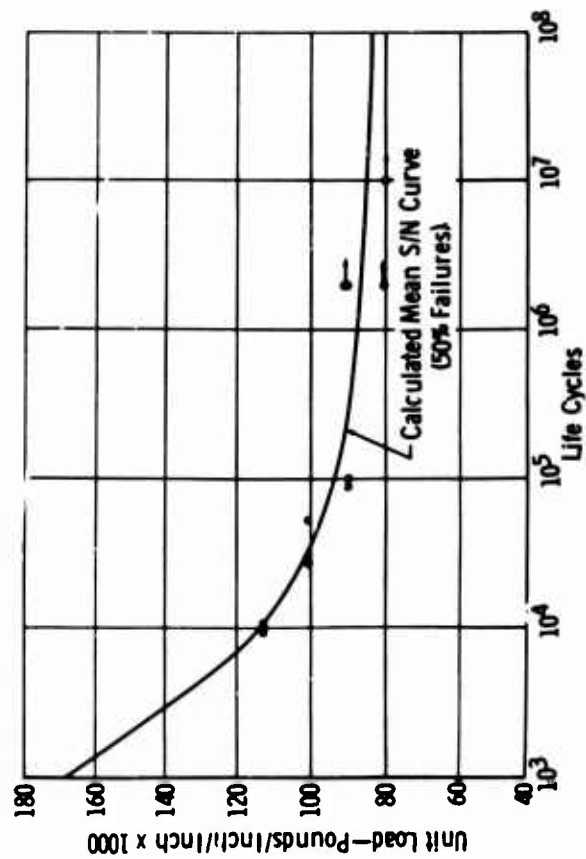


Figure 37. Fatigue Test Results—EX-78786.
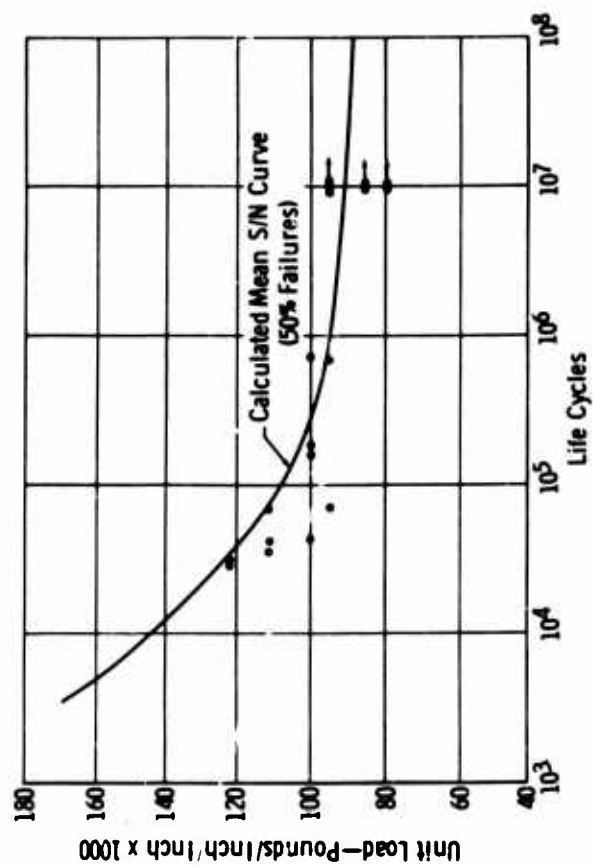


Figure 38. Fatigue Test Results—EX-78787.

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TABLE XI
GEAR TEETH FATIGUE TEST DATA

| Part Number | Serial Number | Tooth Number | Load (pounds) | | | Cycles to Failure | Test Frequency (c. p. s.) | Y Corr. S/N Side |
|---|---|---|---|---|---|---|---|---|
| | | | Static | Dynamic | Total | | | |
| EX-78772 | CX 9092 | 1 | 5340 | 5300 | 10,640 | Void Data | — | 0.3657 |
| | | 2 | 4810 | 4770 | 9,580 | 1 585×10$^4$ | 220 | 0.3657 |
| | | 3 | 4810 | 4770 | 9,580 | 1.715×10$^4$ | 220 | 0.3637 |
| | | 4 | 4810 | 4770 | 9,580 | 2.38×10$^4$ | 220 | 0.3597 |
| | CX 9091 | 1 | 4430 | 4230 | 8,660 | 1.06×10$^4$ | 220 | 0.3697 |
| | | 2 | 4430 | 4230 | 8,660 | 1.32×10$^4$ | 220 | 0.3577 |
| | | 3 | 4430 | 4230 | 8,660 | 1.3×10$^4$ | 220 | 0.3677 |
| | | 4 | 3995 | 3795 | 7,790 | 2.38×10$^4$ | 220 | — |
| | CX 9090 | 1 | 3600 | 3400 | 7,000 | 5.8×10$^4$ | 220 | — |
| | | 2 | 3995 | 3795 | 7,790 | 4.8×10$^4$ | 220 | — |
| | | 3 | 3995 | 3795 | 7,790 | 4.0×10$^4$ | 220 | — |
| EX-78774 | CX 9067 | 1 | 5900 | Void Data High Dynamic Load | | | 220 | 0.3547 |
| | | 2 | 5390 | 5190 | 10,580 | 1.188×10$^4$ | 220 | 0.3607 |
| | | 3 | 5390 | 5190 | 10,580 | 8.9×10$^3$ | 220 | 0.3617 |
| | | 4 | 5390 | 5190 | 10,580 | 6.6×10$^3$ | 220 | 0.3557 |
| | CX 9068 | 1 | 4860 | 4660 | 9,520 | 1.076×10$^4$ | 220 | 0.3576 |
| | | 2 | 4860 | 4660 | 9,520 | 1.32×10$^4$ | 220 | 0.3576 |
| | | 3 | 4860 | 4660 | 9,520 | 1.32×10$^4$ | 220 | 0.3546 |
| | | 4 | 4385 | 4185 | 8,570 | 3.43×10$^4$ | 220 | 0.3586 |
| | CX 9064 | 1 | 4385 | 4185 | 8,570 | 1.32×10$^4$ | 220 | 0.3536 |
| | | 2 | 4385 | 4185 | 8,570 | 1.98×10$^4$ | 220 | 0.3616 |
| | | 3 | 4385 | 4185 | 8,570 | 2.64×10$^4$ | 220 | 0.3536 |
| | | 4 | 4385 | 4185 | 8,570 | 1.85×10$^4$ | 220 | 0.3536 |
| | CX 9065 | 1 | 4385 | 4185 | 8,570 | 1.7×10$^4$ | 220 | 0.3586 |
| | | 2 | 4385 | 4185 | 8,570 | 1.85×10$^4$ | 220 | 0.3606 |
| | | 3 | 4385 | 4185 | 8,570 | 2.64×10$^4$ | 220 | 0.3496 |
| | | 4 | 4385 | 4185 | 8,570 | 1.85×10$^4$ | 220 | 0.3526 |
| EX-78776 | CX 9010 | 1 | 4340 | 4300 | 8,640 | 6.6×10$^3$ | 220 | 0.3793 |
| | | 2 | 3910 | 3870 | 7,780 | 7.92×10$^4$ | 220 | — |
| | | 3 | 3910 | 3870 | 7,780 | 1.32×10$^4$ | 220 | — |
| | | 4 | 3910 | 3870 | 7,780 | 1.04×10$^4$ | 220 | 0.3933 |
| | CX 9008 | 1 | 3600 | 3400 | 7,000 | 1.78×10$^4$ | 220 | — |
| | | 2 | 3600 | 3400 | 7,000 | 5.94×10$^4$ | 220 | 0.3873 |
| | | 3 | 3600 | 3400 | 7,000 | 206×10$^4$ | 220 | — |
| | | 4 | 3250 | 3050 | 6,300 | 6.6×10$^4$ | 220 | 0.3903 |
| | CX 9009 | 1 | 2950 | 2750 | 5,700 | 10$^7$ → | 220 | — |
| | | 2 | 3250 | 3050 | 6,300 | Void Data | — | 0.3883 |
| | | 3 | 3250 | 3050 | 6,300 | Void Data | — | — |
| | | 4 | 3250 | 3050 | 6,300 | 1.3×10$^5$ | 220 | — |
| | CX 9007 | 1 | 3250 | 3050 | 6,300 | 5.3×10$^4$ | 220 | — |
| EX-78778 | CX 9054 | 1 | 4400 | 4200 | 8,600 | 2.9×10$^4$ | 220 | 0.3637 |
| | | 2 | 4400 | 4200 | 8,600 | 3.96×10$^4$ | 220 | 0.3757 |

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A

| Cycles to Failure | Test Frequency (c.p.s.) | Fatigue Crack Dimensions | | | |
|---|---|---|---|---|---|
| | | Y Corrected (inches) | | Z (degrees) | |
| | | S/N Side | Opposite Side | S/N Side | Opposite Side |
| oid Data | — | 0.3657 | 0.3657 | 33 | 36 |
| .585X10$^4$ | 220 | 0.3657 | 0.3657 | 31 | 34 |
| .715X10$^4$ | 220 | 0.3637 | 0.3647 | 27 | 31 |
| .38X10$^4$ | 220 | 0.3597 | 0.3647 | 30 | 31 |
| .06X10$^4$ | 220 | 0.3697 | 0.3697 | 32 | 35 |
| .32X10$^4$ | 220 | 0.3577 | 0.3657 | 30 | 37 |
| .3X10$^4$ | 220 | 0.3677 | 0.3587 | 35 | 30 |
| .38X10$^4$ | 220 | — | — | 32 | — |
| .8X10$^4$ | 220 | — | — | — | — |
| .8X10$^4$ | 220 | — | — | — | — |
| .0X10$^4$ | 220 | — | — | — | — |
| | | | | | |
| Load | 220 | 0.3547 | 0.3637 | 26 | 36 |
| .188X10$^4$ | 220 | 0.3607 | 0.3717 | 32 | 35 |
| .9X10$^3$ | 220 | 0.3617 | 0.3637 | 30 | 32 |
| .6X10$^3$ | 220 | 0.3557 | 0.3617 | 29 | 29 |
| .076X10$^4$ | 220 | 0.3576 | — | 32 | — |
| .32X10$^4$ | 220 | 0.3576 | 0.3596 | 31 | 32 |
| .32X10$^4$ | 220 | 0.3546 | 0.3516 | 31 | 35 |
| .43X10$^4$ | 220 | 0.3586 | 0.3586 | 25 | 32 |
| .32X10$^4$ | 220 | 0.3536 | 0.3536 | 28 | 33 |
| 1.98X10$^4$ | 220 | 0.3616 | 0.3526 | 28 | 34 |
| 2.64X10$^4$ | 220 | 0.3536 | 0.3576 | 33 | 32 |
| 1.85X10$^4$ | 220 | 0.3536 | 0.3576 | 28 | 33 |
| 1.7X10$^4$ | 220 | 0.3586 | 0.3656 | 29 | 28 |
| 1.85X10$^4$ | 220 | 0.3606 | 0.3646 | 30 | 31 |
| 2.64X10$^4$ | 220 | 0.3496 | 0.3616 | 32 | 29 |
| 1.85X10$^4$ | 220 | 0.3526 | 0.3536 | 31 | 29 |
| 5.6X10$^3$ | 220 | 0.3793 | 0.3823 | 26 | 30 |
| 7.92X10$^4$ | 220 | — | — | — | — |
| 1.32X10$^4$ | 220 | — | — | 31 | — |
| 1.04X10$^4$ | 220 | 0.3933 | 0.3973 | 31 | 31 |
| 1.78X10$^4$ | 220 | — | — | 28 | 27 |
| 5.94X10$^4$ | 220 | 0.3873 | 0.3913 | 28 | 29 |
| 206X10$^4$ | 220 | — | — | — | — |
| 6.6X10$^4$ | 220 | 0.3903 | 0.3923 | 28 | 28 |
| 10$^7$ → | 220 | — | — | — | — |
| Void Data | — | 0.3883 | 0.3913 | 21 | 15 |
| Void Data | — | — | — | — | — |
| 1.3X10$^5$ | 220 | — | — | — | — |
| 5.3X10$^4$ | 220 | — | — | — | — |
| | | | | | |
| 2.9X10$^4$ | 220 | 0.3637 | 0.3667 | 28 | 28 |
| 3.96X10$^4$ | 220 | 0.3757 | 0.3847 | 29 | 28 |
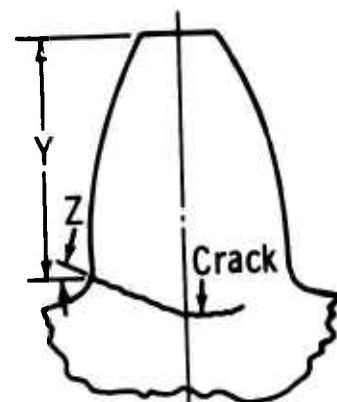


B

TABLE XI (CONT)

| Part Number | Serial Number | Tooth Number | Load (pounds) | | | Cycles to Failure | Test Frequency (c.p.s.) |
|---|---|---|---|---|---|---|---|
| | | | Static | Dynamic | Total | | |
| | | 3 | 4400 | 4200 | 8,600 | $2.1 \times 10^4$ | 220 |
| | | 4 | 3970 | 3770 | 7,740 | $9.23 \times 10^4$ | 220 |
| | CX 9057 | 1 | 3970 | 3770 | 7,740 | $8.71 \times 10^4$ | 220 |
| | | 2 | 3970 | 3770 | 7,740 | $1.346 \times 10^5$ | 220 |
| | | 3 | 3583 | 3383 | 6,965 | Void Data → | — |
| | | 4 | 3583 | 3383 | 6,965 | $2.0 \times 10^6 \rightarrow$ | 220 |
| | CX 9056 | 1 | 3583 | 3383 | 6,965 | $7.65 \times 10^4$ | 220 |
| | | 2 | 3583 | 3383 | 6,965 | $6.6 \times 10^4$ | 220 |
| EX-78780 | CX 9097 | 1 | 4900 | 4700 | 9,600 | $5.28 \times 10^5$ | 220 |
| | | 2 | 4900 | 4700 | 9,600 | $6.6 \times 10^4$ | 220 |
| | | 3 | 4900 | 4700 | 9,600 | $5.94 \times 10^4$ | 220 |
| | | 4 | 5500 | 5300 | 10,800 | $4.62 \times 10^4$ | 220 |
| | CX 9098 | 1 | 5500 | 5300 | 10,800 | $4.125 \times 10^4$ | 220 |
| | CX 9095 | 1 | 5500 | 5300 | 10,800 | $4.62 \times 10^4$ | 220 |
| | | 2 | 4420 | 4220 | 8,640 | $2.0 \times 10^5$ | 220 |
| | | 3 | 4420 | 4220 | 8,640 | $1.85 \times 10^5$ | 220 |
| | | 4 | 4420 | 4220 | 8,640 | $1.85 \times 10^5$ | 220 |
| | CX 9096 | 1 | 6040 | 5840 | 11,880 | $1.32 \times 10^4$ | 220 |
| | | 2 | 6040 | 5840 | 11,880 | $6.6 \times 10^3$ | 220 |
| | | 3 | 6040 | 5840 | 11,880 | $6.6 \times 10^3$ | 220 |
| EX-78782 | CX 9113 | 1 | 6360 | 6160 | 12,520 | $9.5 \times 10^3$ | 220 |
| | | 2 | 6360 | 6160 | 12,520 | Void Data | — |
| | | 3 | 5730 | 5530 | 11,260 | $5.38 \times 10^4$ | 220 |
| | | 4 | 5730 | 5530 | 11,260 | $1.42 \times 10^5$ | 220 |
| | CX 9112 | 1 | 5110 | 4910 | 10,020 | $1.19 \times 10^5$ | 220 |
| | | 2 | 5110 | 4910 | 10,020 | $5.93 \times 10^4$ | 220 |
| | | 3 | 5110 | 4910 | 10,020 | $2.0 \times 10^6 \rightarrow$ | 220 |
| | | 4 | 4600 | 4400 | 9,000 | $10^7 \rightarrow$ | 220 |
| | CX 9111 | 1 | 5730 | 5530 | 11,260 | $1.32 \times 10^4$ | 220 |
| | | 2 | 6360 | 6160 | 12,520 | $1.32 \times 10^4$ | 220 |
| | | 3 | 6360 | 6160 | 12,520 | $1.76 \times 10^4$ | 220 |
| EX-78784 | CX 9072 | 1 | 5250 | 5050 | 10,300 | $1.8 \times 10^4$ | 220 |
| | | 2 | 5250 | 5050 | 10,300 | $1.8 \times 10^4$ | 220 |
| | | 3 | 5250 | 5050 | 10,300 | $8.6 \times 10^3$ | 220 |
| | | 4 | 4220 | 4020 | 8,240 | $1.345 \times 10^5$ | 220 |
| | CX 9070 | 1 | 4220 | 4020 | 8,240 | $2.0 \times 10^6 \rightarrow$ | 220 |
| | | 2 | 4220 | 4020 | 8,240 | $3.313 \times 10^5$ | 220 |
| | | 3 | 3800 | 3600 | 7,400 | $2.0 \times 10^6 \rightarrow$ | 220 |
| | | 4 | 3800 | 3600 | 7,400 | $2.0 \times 10^6 \rightarrow$ | 220 |
| | CX 9073 | 1 | 3800 | 3600 | 7,400 | $3.96 \times 10^3$ | 220 |

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A

| ...ounds) ...mic | Total | Cycles to Failure | Test Frequency (c.p.s.) | Fatigue Crack Dimensions | | | |
|---|---|---|---|---|---|---|---|
| | | | | Y Corrected (inches) | | Z (degrees) | |
| | | | | S/N Side | Opposite Side | S/N Side | Opposite Side |
| | 8,600 | $2.1 \times 10^4$ | 220 | -- | — | — | 28 |
| | 7,740 | $9.23 \times 10^4$ | 220 | 0.3637 | 0.3657 | 30 | 27 |
| | 7,740 | $8.71 \times 10^4$ | 220 | 0.3737 | 0.3767 | 27 | 30 |
| | 7,740 | $1.346 \times 10^5$ | 220 | 0.3587 | 0.3647 | 27 | 29 |
| | 6,965 | Void Data → | — | — | — | — | — |
| | 6,965 | $2.0 \times 10^6$ → | 220 | — | — | — | — |
| | 6,965 | $7.65 \times 10^4$ | 220 | — | — | — | — |
| | 6,965 | $6.6 \times 10^4$ | 220 | — | — | — | — |
| | 9,600 | $5.28 \times 10^5$ | 220 | — | — | 42 | 45 |
| | 9,600 | $6.6 \times 10^4$ | 220 | — | — | — | — |
| | 9,600 | $5.94 \times 10^4$ | 220 | — | — | — | — |
| | 10,800 | $4.62 \times 10^4$ | 220 | 0.3717 | 0.3717 | 38 | 43 |
| | 10,800 | $4.125 \times 10^4$ | 220 | — | — | — | — |
| | 10,800 | $4.62 \times 10^4$ | 220 | — | — | — | — |
| | 8,640 | $2.0 \times 10^5$ | 220 | — | — | — | — |
| | 8,640 | $1.85 \times 10^5$ | 220 | — | — | — | — |
| | 8,640 | $1.85 \times 10^5$ | 220 | — | — | — | — |
| | 11,880 | $1.32 \times 10^4$ | 220 | — | — | — | — |
| | 11,880 | $6.6 \times 10^3$ | 220 | — | — | — | — |
| | 11,880 | $6.6 \times 10^3$ | 220 | — | — | — | — |
| | 12,520 | $9.5 \times 10^3$ | 220 | 0.3599 | 0.3699 | — | — |
| | 12,520 | Void Data | — | 0.3719 | 0.3679 | — | — |
| | 11,260 | $5.38 \times 10^4$ | 220 | 0.3669 | 0.3659 | — | — |
| | 11,260 | $1.42 \times 10^5$ | 220 | — | — | — | — |
| | 10,020 | $1.19 \times 10^5$ | 220 | — | — | — | — |
| | 10,020 | $5.93 \times 10^4$ | 220 | 0.3628 | 0.3688 | 39 | 41 |
| | 10,020 | $2.0 \times 10^6$ → | 220 | — | — | — | — |
| | 9,000 | $10^7$ → | 220 | — | — | — | — |
| | 11,260 | $1.32 \times 10^4$ | 220 | — | — | — | — |
| | 12,520 | $1.32 \times 10^4$ | 220 | — | — | — | — |
| | 12,520 | $1.76 \times 10^4$ | 220 | — | — | — | — |
| | 10,300 | $1.8 \times 10^4$ | 220 | 0.3731 | 0.3921 | 34 | 32 |
| | 10,300 | $1.8 \times 10^4$ | 220 | 0.3911 | 0.3941 | 31 | 30 |
| | 10,300 | $8.6 \times 10^3$ | 220 | 0.3901 | 0.3941 | 35 | 36 |
| | 8,240 | $1.345 \times 10^5$ | 220 | 0.3961 | 0.3981 | 31 | 39 |
| | 8,240 | $2.0 \times 10^6$ → | 220 | — | — | — | — |
| | 8,240 | $3.313 \times 10^5$ | 220 | 0.3869 | 0.3919 | 34 | 34 |
| | 7,400 | $2.0 \times 10^6$ → | 220 | — | — | — | — |
| | 7,400 | $2.0 \times 10^6$ → | 220 | — | — | — | — |
| | 7,400 | $3.96 \times 10^3$ | 220 | 0.3881 | 0.3951 | 35 | 42 |

B

TABLE XI (CONT)

| Part Number | Serial Number | Tooth Number | Load (pounds) | | | Cycles to Failure | Test Frequency (c.p.s.) | Y Corre... S/N Side |
|---|---|---|---|---|---|---|---|---|
| | | | Static | Dynamic | Total | | | |
| | | 2 | 3800 | 3600 | 7,400 | $8.58 \times 10^4$ | 220 | 0.3821 |
| | | 3 | 3800 | 3600 | 7,400 | $7.1 \times 10^4$ | 220 | — |
| | | 4 | 4735 | 4535 | 9,270 | $1.76 \times 10^4$ | 220 | 0.3881 |
| | CX 9071 | 1 | 4735 | 4535 | 9,270 | $3.16 \times 10^4$ | 220 | — |
| | | 2 | 4735 | 4535 | 9,270 | Void Data | — | — |
| | | 3 | 4735 | 4535 | 9,270 | $1.85 \times 10^4$ | 220 | — |
| EX-78786 | CX 9013 | 1 | 5295 | 5095 | 10,390 | $1.057 \times 10^4$ | 220 | 0.3842 |
| | | 2 | 5295 | 5095 | 10,390 | $9.23 \times 10^3$ | 220 | 0.3862 |
| | | 3 | 5295 | 5095 | 10,390 | $9.9 \times 10^3$ | 220 | 0.3872 |
| | | 4 | 4260 | 4060 | 8,320 | $9.77 \times 10^4$ | 220 | — |
| | CX 9014 | 1 | 4260 | 4060 | 8,320 | $2 \times 10^6 \rightarrow$ | 220 | — |
| | | 2 | 4260 | 4060 | 8,320 | $2 \times 10^6 \rightarrow$ | 220 | — |
| | | 3 | 3830 | 3660 | 7,490 | $2 \times 10^6 \rightarrow$ | 220 | — |
| | | 4 | 3830 | 3660 | 7,490 | $2 \times 10^6 \rightarrow 10^7 \rightarrow$ | 220 | — |
| | CX 9015 | 1 | 4775 | 4575 | 9,350 | $2.64 \times 10^4$ | 220 | — |
| | | 2 | 4775 | 4575 | 9,350 | $2.64 \times 10^4$ | 220 | — |
| | | 3 | 4775 | 4575 | 9,350 | $5.28 \times 10^4$ | 220 | — |
| | | 4 | 4260 | 4060 | 8,320 | $9.2 \times 10^4$ | 220 | 0.3822 |
| EX-78773 | CX 9076 | 1 | 678 | 658 | 1,335 | $2.0 \times 10^6 \rightarrow$ | 240 | — |
| | | 2 | 1198 | 1178 | 2,375 | $1.0 \times 10^5$ | 240 | — |
| | | 3 | 1198 | 1178 | 2,375 | $1.58 \times 10^5$ | 240 | — |
| | | 4 | 1198 | 1178 | 2,375 | $4.32 \times 10^4$ | 240 | — |
| | CX 9077 | 1 | 1303 | 1283 | 2,585 | $2.1 \times 10^4$ | 50 | 0.1830 |
| | | 2 | 1303 | 1283 | 2,585 | $2.4 \times 10^4$ | 50 | 0.1860 |
| | | 3 | 1303 | 1283 | 2,585 | $1.5 \times 10^4$ | 50 | 0.1830 |
| | | 4 | 1073 | 1053 | 2,125 | $2.0 \times 10^6 \rightarrow$ | 240 | 0.1870 |
| | CX 9075 | 1 | 1073 | 1053 | 2,125 | $1.29 \times 10^4$ | 240 | 0.1849 |
| | | 2 | 1073 | 1053 | 2,125 | $5.04 \times 10^4$ | 240 | — |
| | | 3 | 1073 | 1053 | 2,125 | $2.88 \times 10^4$ | 240 | — |
| | | 4 | 1198 | 1178 | 2,375 | $3.96 \times 10^4$ | 240 | — |
| | CX 9074 | 1 | 1198 | 1178 | 2,375 | $2.11 \times 10^4$ | 240 | 0.1829 |
| | | 2 | 1198 | 1178 | 2,375 | $1.85 \times 10^4$ | 240 | — |
| | | 3 | 966 | 946 | 1,912 | $1.05 \times 10^5$ | 240 | — |
| | | 4 | 966 | 946 | 1,912 | $2 \times 10^6 \rightarrow$ | 240 | — |
| | CX 9078 | 1 | 966 | 946 | 1,912 | $3.16 \times 10^4$ | 240 | 0.1829 |
| EX-78775 | CX 9099 | 1 | 1135 | 1115 | 2,250 | $2.0 \times 10^6 \rightarrow$ | 240 | — |
| | | 2 | 1198 | 1178 | 2,375 | $2.0 \times 10^7 \rightarrow$ | 240 | — |
| | | 3 | 1303 | 1283 | 2,585 | $1.296 \times 10^5$ | 240 | — |
| | | 4 | 1303 | 1283 | 2,585 | $3.6 \times 10^4$ | 240 | 0.1675 |

A

| l (pounds) | | Cycles to Failure | Test Frequency (c. p. s.) | Fatigue Crack Dimensions | | | |
| ynamic | Total | | | Y Corrected (inches) | | Z (degrees) | |
| | | | | S/N Side | Opposite Side | S/N Side | Opposite Side |
|---|---|---|---|---|---|---|---|
| 3600 | 7,400 | $8.58 \times 10^4$ | 220 | 0.3821 | 0.3891 | 34 | 36 |
| 3600 | 7,400 | $7.1 \times 10^4$ | 220 | — | — | — | — |
| 4535 | 9,270 | $1.76 \times 10^4$ | 220 | 0.3881 | 0.3911 | 36 | 38 |
| 4535 | 9,270 | $3.16 \times 10^4$ | 220 | — | — | — | — |
| 4535 | 9,270 | Void Data | — | — | — | — | — |
| 4535 | 9,270 | $1.85 \times 10^4$ | 220 | — | — | — | — |
| 5095 | 10,390 | $1.057 \times 10^4$ | 220 | 0.3842 | 0.3882 | 36 | 35 |
| 5095 | 10,390 | $9.23 \times 10^3$ | 220 | 0.3862 | 0.3872 | 33 | 33 |
| 5095 | 10,390 | $9.9 \times 10^3$ | 220 | 0.3872 | 0.3932 | 33 | 32 |
| 4060 | 8,320 | $9.77 \times 10^4$ | 220 | — | — | — | — |
| 4060 | 8,320 | $2 \times 10^6 \rightarrow$ | 220 | — | — | — | — |
| 4060 | 8,320 | $2 \times 10^6 \rightarrow$ | 220 | — | — | — | — |
| 3660 | 7,490 | $2 \times 10^6 \rightarrow$ | 220 | — | — | — | — |
| 3660 | 7,490 | $2 \times 10^6 \rightarrow 10^7 \rightarrow$ | 220 | — | — | — | — |
| 4575 | 9,350 | $2.64 \times 10^4$ | 220 | — | — | 30 | — |
| 4575 | 9,350 | $2.64 \times 10^4$ | 220 | — | — | — | — |
| 4575 | 9,350 | $5.28 \times 10^4$ | 220 | — | — | — | — |
| 4060 | 8,320 | $9.2 \times 10^4$ | 220 | 0.3822 | 0.3852 | 30 | 36 |
| 658 | 1,335 | $2.0 \times 10^6 \rightarrow$ | 240 | — | — | — | — |
| 178 | 2,375 | $1.0 \times 10^5$ | 240 | — | — | — | — |
| 178 | 2,375 | $1.58 \times 10^5$ | 240 | — | — | — | — |
| 178 | 2,375 | $4.32 \times 10^4$ | 240 | — | — | — | — |
| 283 | 2,585 | $2.1 \times 10^4$ | 50 | 0.1830 | 0.1880 | 31 | 38 |
| 283 | 2,585 | $2.4 \times 10^4$ | 50 | 0.1860 | 0.1830 | 31 | 36 |
| 283 | 2,585 | $1.5 \times 10^4$ | 50 | 0.1830 | 0.1830 | 30 | 35 |
| 053 | 2,125 | $2.0 \times 10^6 \rightarrow$ | 240 | 0.1870 | 0.1800 | 33 | 26 |
| 053 | 2,125 | $1.29 \times 10^4$ | 240 | 0.1849 | 0.1859 | 28 | 37 |
| 053 | 2,125 | $5.04 \times 10^4$ | 240 | — | — | — | — |
| 053 | 2,125 | $2.88 \times 10^4$ | 240 | — | — | — | — |
| 178 | 2,375 | $3.96 \times 10^4$ | 240 | — | — | — | — |
| 178 | 2,375 | $2.11 \times 10^4$ | 240 | 0.1829 | 0.1849 | 29 | 32 |
| 178 | 2,375 | $1.85 \times 10^4$ | 240 | — | — | — | — |
| 946 | 1,912 | $1.05 \times 10^5$ | 240 | — | — | — | — |
| 946 | 1,912 | $2 \times 10^6 \rightarrow$ | 240 | — | — | — | — |
| 946 | 1,912 | $3.16 \times 10^4$ | 240 | 0.1829 | 0.1809 | 30 | 31 |
| 115 | 2,250 | $2.0 \times 10^6 \rightarrow$ | 240 | — | — | — | — |
| 178 | 2,375 | $2.0 \times 10^7 \rightarrow$ | 240 | — | — | — | — |
| 283 | 2,585 | $1.296 \times 10^5$ | 240 | — | — | — | — |
| 283 | 2,585 | $3.6 \times 10^4$ | 240 | 0.1675 | 0.1715 | — | — |

B

TABLE XI (CONT)

| Part Number | Serial Number | Tooth Number | Load (pounds) | | | Cycles to Failure | Test Frequency (c. p. s.) |
|---|---|---|---|---|---|---|---|
| | | | Static | Dynamic | Total | | |
| | CX 9033 | 1 | 1460 | 1440 | 2,900 | $2.4\times10^4$ | 50 |
| | | 2 | 1605 | 1585 | 3,190 | $1.8\times10^4$ | 50 |
| | | 3 | 1605 | 1585 | 3,190 | $2.1\times10^4$ | 50 |
| | | 4 | 1765 | 1745 | 3,510 | $1.65\times10^4$ | 50 |
| | CX 9034 | 1 | 1160 | 1140 | 2,300 | $2\times10^6 \rightarrow$ | 240 |
| | | 2 | 1160 | 1140 | 2,300 | $2\times10^6 \rightarrow$ | 240 |
| | | 3 | 1330 | 1310 | 2,640 | $2\times10^6 \rightarrow$ | 240 |
| | | 4 | 1330 | 1310 | 2,640 | $2\times10^6 \rightarrow$ | 240 |
| EX-78783 | CX 9025 | 1 | 1160 | 1140 | 2,300 | $2\times10^6 \rightarrow$ | 240 |
| | | 2 | 1160 | 1140 | 2,300 | $2\times10^6 \rightarrow$ | 240 |
| | | 3 | 1330 | 1310 | 2,640 | $1.73\times10^5$ | 240 |
| | | 4 | 1330 | 1310 | 2,640 | $4.03\times10^5$ | 240 |
| | CX 9026 | 1 | 1460 | 1440 | 2,900 | $2.0\times10^6 \rightarrow$ | 240 |
| | | 2 | 1460 | 1440 | 2,900 | $1.008\times10^5$ | 240 |
| | | 3 | 1510 | 1490 | 3,000 | $2.52\times10^4$ | 50 |
| | | 4 | 1510 | 1490 | 3,000 | $1.98\times10^4$ | 50 |
| | CX 9027 | 1 | 1510 | 1490 | 3,000 | $4.32\times10^4$ | 50 |
| | | 2 | 1660 | 1640 | 3,300 | $1.95\times10^4$ | 50 |
| | | 3 | 1660 | 1640 | 3,300 | $1.5\times10^4$ | 50 |
| | | 4 | 1660 | 1640 | 3,300 | $2.55\times10^4$ | 50 |
| | CX 9028 | 1 | 1810 | 1790 | 3,600 | $1.44\times10^4$ | 50 |
| | | 2 | 1810 | 1790 | 3,600 | $1.53\times10^4$ | 50 |
| | | 3 | 1810 | 1790 | 3,600 | $7.5\times10^3$ | 50 |
| | CX 9029 | 1 | 1460 | 1440 | 2,900 | $2.68\times10^5$ | 240 |
| | | 2 | 1460 | 1440 | 2,900 | $5.76\times10^5$ | 240 |
| | | 3 | 1330 | 1310 | 2,640 | $7.2\times10^3$ | 50 |
| | | 4 | 1330 | 1310 | 2,640 | $2.1\times10^4$ | 50 |
| EX-78785 | CX 9035 | 1 | 1200 | 1160 | 2,360 | $1.15\times10^5$ | 240 |
| | | 2 | 950 | 928 | 1,878 | $3.6\times10^4$ | 240 |
| | | 3 | 850 | 800 | 1,650 | $10^7 \rightarrow$ | 240 |
| | | 4 | 890 | 860 | 1.750 | $10^7 \rightarrow$ | 240 |
| | CX 9037 | 1 | 1100 | 1080 | 2,180 | $4.32\times10^4$ | 50 |
| | | 2 | 1100 | 1080 | 2,180 | $5.04\times10^4$ | 50 |
| | | 3 | 1040 | 1020 | 2,060 | $1.29\times10^5$ | 50 |
| | | 4 | 1040 | 1020 | 2,060 | $1.512\times10^5$ | 50 |
| | CX 9038 | 1 | 1160 | 1140 | 2,300 | $9.37\times10^4$ | 50 |
| | | 2 | 1160 | 1140 | 2,300 | $4.5\times10^4$ | 50 |
| | | 3 | 1160 | 1140 | 2,300 | $1.62\times10^4$ | 50 |
| | | 4 | 1100 | 1080 | 2,180 | $2.16\times10^4$ | 50 |

A

| Y S/N | Load (pounds) | | Cycles to Failure | Test Frequency (c.p.s.) | Fatigue Crack Dimensions | | | |
|---|---|---|---|---|---|---|---|---|
| | Dynamic | Total | | | Y Corrected (inches) | | Z (degrees) | |
| | | | | | S/N Side | Opposite Side | S/N Side | Opposite Side |
| 0. | 1440 | 2,900 | $2.4 \times 10^4$ | 50 | 0.1769 | 0.1769 | 31 | 33 |
| 0. | 1585 | 3,190 | $1.8 \times 10^4$ | 50 | 0.1789 | 0.1769 | 34 | 34 |
| 0. | 1585 | 3,190 | $2.1 \times 10^4$ | 50 | 0.1789 | 0.1789 | 32 | 37 |
| 0. | 1745 | 3,510 | $1.65 \times 10^4$ | 50 | 0.1759 | 0.1759 | 31 | 36 |
| 0. | 1140 | 2,300 | $2 \times 10^6 \rightarrow$ | 240 | — | — | — | — |
| | 1140 | 2,300 | $2 \times 10^6 \rightarrow$ | 240 | — | — | — | — |
| | 1310 | 2,640 | $2 \times 10^6 \rightarrow$ | 240 | — | — | — | — |
| | 1310 | 2,640 | $2 \times 10^6 \rightarrow$ | 240 | — | — | — | — |
| | 1140 | 2,300 | $2 \times 10^6 \rightarrow$ | 240 | — | — | — | — |
| | 1140 | 2,300 | $2 \times 10^6 \rightarrow$ | 240 | — | — | — | — |
| | 1310 | 2,640 | $1.73 \times 10^5$ | 240 | — | — | — | — |
| | 1310 | 2,640 | $4.00 \times 10^5$ | 240 | — | — | — | — |
| | 1440 | 2,900 | $2.0 \times 10^6 \rightarrow$ | 240 | — | — | — | — |
| | 1440 | 2,900 | $1.008 \times 10^5$ | 240 | — | — | — | — |
| 0. | 1490 | 3,000 | $2.52 \times 10^4$ | 50 | 0.1807 | 0.1857 | 32 | 41 |
| 0. | 1490 | 3,000 | $1.98 \times 10^4$ | 50 | 0.1847 | 0.1847 | 36 | 37 |
| | 1490 | 3,000 | $4.32 \times 10^4$ | 50 | — | — | — | — |
| 0. | 1640 | 3,300 | $1.95 \times 10^4$ | 50 | 0.1867 | 0.1827 | 35 | 36 |
| 0. | 1640 | 3,300 | $1.5 \times 10^4$ | 50 | 0.1787 | 0.1807 | 34 | 36 |
| | 1640 | 3,300 | $2.55 \times 10^4$ | 50 | — | — | — | — |
| | 1790 | 3,600 | $1.44 \times 10^4$ | 50 | — | — | — | — |
| | 1790 | 3,600 | $1.53 \times 10^4$ | 50 | — | — | — | — |
| | 1790 | 3,600 | $7.5 \times 10^3$ | 50 | — | — | — | — |
| 0. | 1440 | 2,900 | $2.68 \times 10^5$ | 240 | 0.1720 | 0.1740 | 31 | 33 |
| | 1440 | 2,900 | $5.76 \times 10^5$ | 240 | — | — | — | — |
| | 1310 | 2,640 | $7.2 \times 10^3$ | 50 | — | — | — | — |
| | 1310 | 2,640 | $2.1 \times 10^4$ | 50 | — | — | — | — |
| 0. | 1160 | 2,360 | $1.15 \times 10^5$ | 240 | 0.1891 | 0.1871 | 28 | 31 |
| | 928 | 1,878 | $3.6 \times 10^4$ | 240 | — | — | — | — |
| | 800 | 1,650 | $10^7 \rightarrow$ | 240 | — | — | — | — |
| | 860 | 1,750 | $10^7 \rightarrow$ | 240 | — | — | — | — |
| | 1080 | 2,180 | $4.32 \times 10^4$ | 50 | — | — | — | — |
| | 1080 | 2,180 | $5.04 \times 10^4$ | 50 | — | — | — | — |
| | 1020 | 2,060 | $1.29 \times 10^5$ | 50 | — | — | — | — |
| | 1020 | 2,060 | $1.512 \times 10^5$ | 50 | — | — | — | — |
| 0. | 1140 | 2,300 | $9.37 \times 10^4$ | 50 | 0.1901 | 0.1921 | 26 | 32 |
| | 1140 | 2,300 | $4.5 \times 10^4$ | 50 | — | — | — | — |
| | 1140 | 2,300 | $1.62 \times 10^4$ | 50 | — | — | — | — |
| | 1080 | 2,180 | $2.16 \times 10^4$ | 50 | — | — | — | — |

β

TABLE XI (CONT)

| Part Number | Serial Number | Tooth Number | Load (pounds) | | | Cycles to Failure | Test Frequency (c. p. s.) | Y ... S/N |
|---|---|---|---|---|---|---|---|---|
| | | | Static | Dynamic | Total | | | |
| EX-78787 | CX 9114 | 1 | 1160 | 1140 | 2,300 | $4.32 \times 10^4$ | 240 | |
| | | 2 | 1160 | 1140 | 2,300 | $1.87 \times 10^5$ | 240 | |
| | | 3 | 1160 | 1140 | 2,300 | $7.2 \times 10^5$ | 240 | |
| | CX 9115 | 1 | 1100 | 1080 | 2,180 | $10^7 \rightarrow$ | 240 | |
| | | 2 | 1100 | 1080 | 2,180 | $6.91 \times 10^5$ | 240 | |
| | | 3 | 1100 | 1080 | 2,180 | $10^7 \rightarrow$ | 240 | |
| | | 4 | 1100 | 1080 | 2,180 | $10^7 \rightarrow$ | 240 | |
| | CX 9116 | 1 | 1285 | 1265 | 2,550 | $6.9 \times 10^4$ | 50 | 0.1 |
| | | 2 | 1285 | 1265 | 2,550 | $4.2 \times 10^4$ | 50 | 0.1 |
| | | 3 | 1285 | 1265 | 2,550 | $3.6 \times 10^4$ | 50 | |
| | | 4 | 1415 | 1395 | 2,810 | $2.85 \times 10^4$ | 50 | |
| | CX 9117 | 1 | 935 | 915 | 1,850 | $10^7 \rightarrow$ | 240 | |
| | | 2 | 935 | 915 | 1,850 | $10^7 \rightarrow$ | 240 | |
| | | 3 | 980 | 970 | 1,950 | $10^7 \rightarrow$ | 240 | |
| | | 4 | 980 | 970 | 1,950 | $10^7 \rightarrow$ | 240 | |
| | CX 9118 | 1 | 1415 | 1395 | 2,810 | $3 \times 10^4$ | 50 | 0.1 |
| | | 2 | 1415 | 1395 | 2,810 | $2.94 \times 10^5$ | 50 | |

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A

| ...unds) ..mic | Total | Cycles to Failure | Test Frequency (c.p.s.) | Fatigue Crack Dimensions | | | |
|---|---|---|---|---|---|---|---|
| | | | | Y Corrected (inches) | | Z (degrees) | |
| | | | | S/N Side | Opposite Side | S/N Side | Opposite Side |
| | 2,300 | $4.32 \times 10^4$ | 240 | — | — | 26 | 27 |
| | 2,300 | $1.87 \times 10^5$ | 240 | — | — | — | — |
| | 2,300 | $7.2 \times 10^5$ | 240 | — | — | — | — |
| | 2,180 | $10^7 \rightarrow$ | 240 | — | — | — | — |
| | 2,180 | $6.91 \times 10^5$ | 240 | — | — | — | — |
| | 2,180 | $10^7 \rightarrow$ | 240 | — | — | — | — |
| | 2,180 | $10^7 \rightarrow$ | 240 | — | — | — | — |
| | 2,550 | $6.9 \times 10^4$ | 50 | 0.1890 | 0.1920 | 27 | 32 |
| | 2,550 | $4.2 \times 10^4$ | 50 | 0.1890 | 0.1930 | 27 | 32 |
| | 2,550 | $3.6 \times 10^4$ | 50 | — | — | — | — |
| | 2,810 | $2.85 \times 10^4$ | 50 | — | — | — | — |
| | 1,850 | $10^7 \rightarrow$ | 240 | — | — | — | — |
| | 1,850 | $10^7 \rightarrow$ | 240 | — | — | — | —. |
| | 1,950 | $10^7 \rightarrow$ | 240 | — | — | — | — |
| | 1,950 | $10^7 \rightarrow$ | 240 | — | — | — | — |
| | 2,810 | $3 \times 10^4$ | 50 | 0.1843 | 0.1893 | 25 | 34 |
| | 2,810 | $2.94 \times 10^5$ | 50 | — | — | — | — |

B

rework involved fabricating tips with carburized surfaces. The carburized surfaces did not distort under high load; thus, carburizing appeared to be a desirable process for this type of testing. It is believed that the difficulties encountered did not affect the data because each condition was recognized early and was corrected.

Another difficulty involved tip rotation under high loads during the fatigue test of the 4.0-inch-pitch-diameter gears. By rotating, the load point was changed; thus, one data point was affected and was discarded. To prevent rotation, a small piece of shim stock was spot-welded to the outside diameter of the tip and load cell, locking the two together and preventing rotation.

### Gage Locating Block

Interference between the gage locating block and the stub tooth was discovered early in the program. This interference would have prevented true angular positioning of the gear tooth on the contact surface of the tip, thus defining a load point other than the high point of single tooth contact. The gage blocks were reworked for clearance; no data points were affected.

### Bias Spring

The original bias spring had a spring rate of 2000 pounds per inch, which was not sufficient to preload the 4.0-inch-pitch-diameter gears. Therefore, springs with a spring rate of 20,000 pounds per inch were purchased to satisfy the preload requirements.

### Load Cell

It was discovered during the rework of the tips that the squareness and flatness of the tip surface mating with the load cell affected load cell calibration. The rework that most effectively corrected this difficulty was lapping of the two surfaces. Once good surface contact was established, the difficulty was eliminated. A number of data points (32 total) were affected by this condition. A series of tests was conducted where this condition existed; the test was duplicated. This yielded a correction factor which was applied to the affected data points. It is believed that the data were corrected with sufficient accuracy to avoid distortion of the final evaluation.

### Test Frequency

The gears having a diametral pitch of 12 were tested at two frequencies—50 and 240 c.p.s. The frequency of 240 c.p.s. was at system resonance. The 50-c.p.s. frequency was selected for use at the higher test loads to provide increased duration of fatigue test time. The time required to establish the test rig load was thereby maintained small when compared with the fatigue time at load. The literature indicates that less than a 2-percent difference in fatigue life would be expected from this change in frequency (reference 20). A similar nonresonance operating procedure was not possible with the gears having a diametral pitch of 6 without overloading the shaker. Quicker establishment of the load on the larger gears was possible without overloading, so there was no strong requirement for a drop in test frequency.

### FAILED GEAR TOOTH CRACK MEASUREMENTS

A comparison was made of the calculated location of the weakest section of each

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tooth and the actual location. To do this, the crack in each failed tooth was measured and recorded. See Table XI. The bar charts in Figures 39 and 40 summarize the results of this investigation. For each configuration, the location of the crack at the tooth surface was measured from the outside diameter and center line of the tooth, within an estimated 0.002 inch. The average dimension corrected for outside diameter variations is plotted for comparison with the theoretical locations as determined by both Lewis and Kelley-Pedersen construction. The charts indicate that for all configurations, Kelley-Pedersen construction locates the weakest section of the tooth closer to the actual measured location than does Lewis construction. The gears having a diametral pitch of 12 show the measured location to be, on the average, 0.015 inch closer to the root than the Lewis theoretical locations. In the gears having a diametral pitch of 6, the deviation is proportional or 0.030 inch closer to the root than the calculated Lewis location. For a graphical presentation of these data, a typical tooth profile trace of each configuration was made. Two such traces are shown in Figures 41 and 42. The weakest section is shown on each trace as calculated by Lewis and Kelley-Pedersen and as measured.

It would be natural to conclude from the examination of these results alone that the Kelley-Pedersen construction provides a more accurate means to locate the true weakest section of the tooth. However, fatigue test data have already shown that the AGMA stress formula using the Lewis tooth form factor most nearly approximates the endurance characteristics of the gear material. The reason for this paradox may be the change of tooth geometry as the tooth deflects under load. Another possibility is the Kelley-Pedersen stress formula, which was derived from a photoelastic study. It may be assumed that the method derived for locating the weakest section is accurate, as the experimental data show. However, the stress concentration factor employed may require modification to obtain a stress value comparable to the true stress in the material. Unfortunately, further pursuit of this phase of the investigation was not possible within the scope of this program; it should be considered, however, in future studies.

Crack measurements were obtained on twelve EX-78774 gears (configuration 3). These data were statistically analyzed to calculate a standard deviation of $0.48 \times 10^{-4}$ and a variance of $0.234 \times 10^{-4}$ from the 0.3581 corrected average "Y" value for this configuration. These data tend to indicate the consistency of fatigue test gear manufacturing and test.

## METALLURGICAL INVESTIGATIONS

Metallurgical examinations of failed test gears were conducted to determine mode of failure, origin of failure, microstructure, case depth, hardness gradient, and material cleanliness.

Six gears were submitted for metallurgical investigation as follows:

| Part Number | Serial Number |
|-------------|---------------|
| EX-78773 | CX 9077 |
| EX-78775 | CX 9100 |
| EX-78777 | CX 9059 |
| EX-78779 | CX 9104 |
| EX-78782 | CX 9113 |
| EX-78784 | CX 9069 |

Code: ▬▬ Measured Location
▪▪▪▪▪▪ Lewis Calculation
▪▪▪▪ Kelley-Pedersen

Configuration

1—20° PA,
0.050 R Ground
Fillet

3—20° PA,
0.080 R Ground
Fillet

5—20° PA,
0.050 R Protub-
erant Fillet

7—20° PA,
0.080 R Protub-
erant Fillet

9—25° PA, 0.050 R
Ground Fillet

11—25° PA,
0.067 R Ground
Fillet

13—25° PA,
0.050 R Protub-
erant Fillet

15—25° PA,
0.067 R Protub-
erant Fillet

**Figure 40.** Location of Fracture Compared With Calculated Location of Weakest Section From Gear Outside Diameter (Diametral Pitch = 12).



Code: ▬▬ Measured Location
▪▪▪▪▪▪ Lewis Calculation
▪▪▪▪ Kelley-Pedersen

Configuration

2—20° PA,
0.025 R Ground
Fillet

4—20° PA,
0.040 R Ground
Fillet

6—20° PA,
0.025 R Protub-
erant Fillet

8—20° PA,
0.040 R Protub-
erant Fillet

10—25° PA,
0.025 R
Ground Fillet

12—25° PA,
0.033 R Ground
Fillet

14—25° PA,
0.025 R Protub-
erant Fillet

16—25° PA,
0.033 R Protub-
erant Fillet

**Figure 39.** Location of Fracture Compared With Calculated Location of Weakest Section From Gear Outside Diameter (Diametral Pitch = 6).

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Part No.  EX-78772 (No. 1)
Serial No. CX9092
Tooth No. 1

Load Side of Tooth

Figure 41.  Typical Tooth Profile
Trace—EX-78772.

Lewis

Calculated (Kelley-Pedersen)

Origin of
Failure

Actual

Part No.  EX-78776 (No. 5)
Serial No. CX9008
Tooth No. 2

Load Side of Tooth
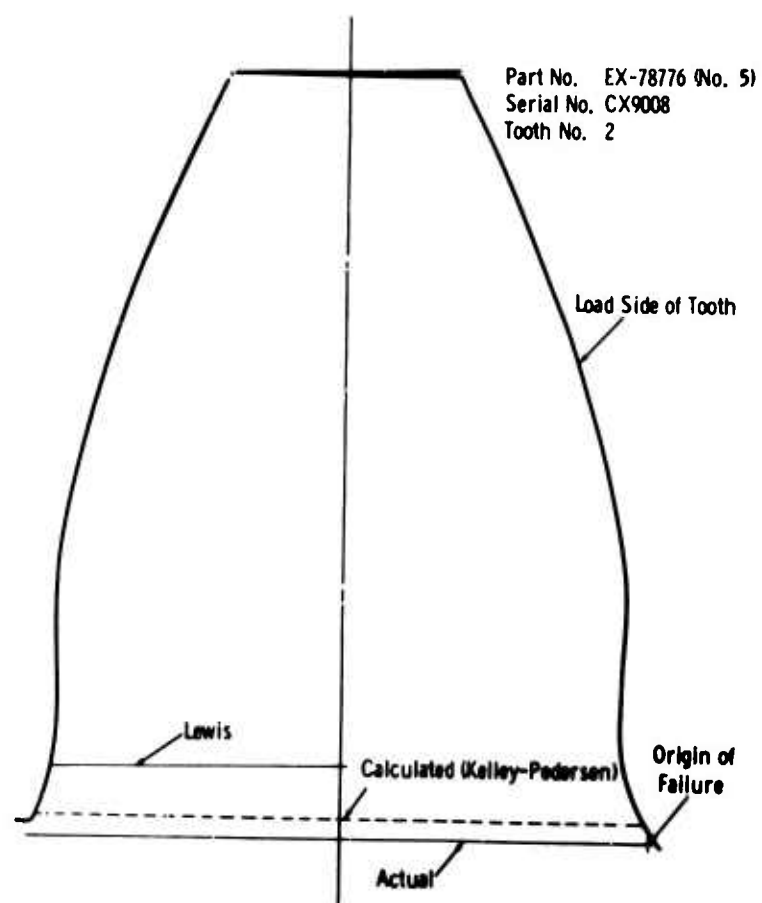
Figure 42.  Typical Tooth Profile
Trace—EX-78776.

Lewis

Calculated (Kelley-Pedersen)

Origin of
Failure

Actual

The following metallurgical conclusions were made.

- Failure of the tested teeth occurred in fatigue.
- The fatigue failures of the tested gear teeth originated in the carburized case of the root radius below the loaded involute.
- Electron fractographs were used to determine the precise origin of failure. The failures appeared to be predominantly multiple.
- The microstructure of the carburized case of the various gears was typical of spheroidized carbides in a martensitic matrix with no indication of carbide network in the areas of failure in the root radii. The core microstructures were of tempered martensite.
- The effective case depth, measured to the $R_c$ 50 level, was indicated to be approximately 0.030 inch on test gears (EX-78773 and EX-78775); approximately 0.040 inch on test gears EX-78777, EX-78779, and EX-78782; and approximately 0.050 inch on test gear EX-78784.
- The test gear material was clean and free from inclusions.
- The material conformed to the compositional requirements of AMS-6265.

Electron fractographs of the failure surfaces of the four failed teeth of test gear EX-78784, serial number CX 9069, confirmed a fatigue failure mode on each surface, as shown in Figures 43, 44, 45, and 46. Visual examination of the failure surfaces of the failed teeth of all submitted gears revealed similar straight-line failures, some of which displayed occasional arrest lines of progressing, typical of fatigue, originating in the root radii. Visual examination of test gear EX-78782, serial number CX 9113, revealed an additional fatigue failure progressing radially from below the root on the nonloaded side of a failed tooth to the center of the gear. (This isolated failure, discussed in the subsection titled Fatigue Tests, was due to localized temperature and was subsequently corrected by cooling the gear.) Microexamination of transverse sections through the failure surfaces of failed teeth from each of the submitted gears revealed straight-line failures typical of fatigue. These failures originated in the carburized case structure in the root radius below the loaded involute, as shown in Figures 47 through 52. The failures, typically, had multiple origins, indicating equalized loading in clean material. Unetched, polished specimens revealed good material quality. The microstructures were of spheroidized carbides in a martensitic matrix with no carbide network in the case and tempered martensite in the core. A typical core microstructure of tempered martensite is shown in Figure 53. Effective case depth measured to the $R_c$ 50 level varied approximately 0.030 inch on part numbers EX-78773 and EX-78775; approximately 0.040 inch on part numbers EX-78777, EX-78779, and EX-78782; and approximately 0.050 inch on part number EX-78784. Case hardness of the various test gears was $R_c$ 61 to 62 at 0.002 inch below the surface with a diminishing gradient as shown in Table XII. Spectrographic analysis indicated conformance of the material in the test gears to the compositional requirements of AMS-6265. Photographs indicating case depths around root fillet contour are shown in Figures 54 through 59.

Fluorescent penetrant inspection of the test gears indicated that all failures of the teeth occurred in the root radii, as indicated in Figures 60 through 65. Fluorescent penetrant inspection of test gear part number EX-78782, serial number CX 9113, revealed an additional radial crack, as shown in Figure 64. Visual examination of the surfaces of failure revealed flat fractures with multiple origins of failure, but only occasional arrest lines indicative of fatigue, as shown in Figures 66 through 70. Visual examination of the failure surface of the radial failure in test gear part number EX-78782, serial number CX 9113, revealed a smooth failure with arrest lines of progression, typical of fatigue, originating below the root radius on the unloaded side of a failed tooth and progressing to the hub, as shown in Figure 71.
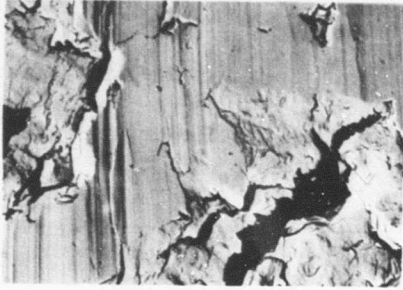
Magnification: 2,500×          Magnification: 10,000×

EX-78784, Serial Number CX 9069

Figure 43.   Fractographs of Surface of Failure of Gear Tooth Number 1 Showing
Failure Contour Typical of Fatigue.
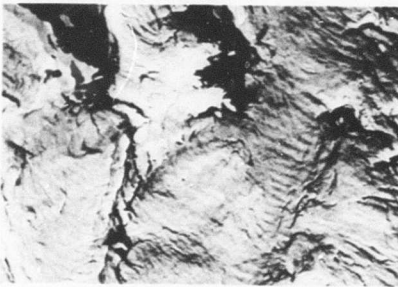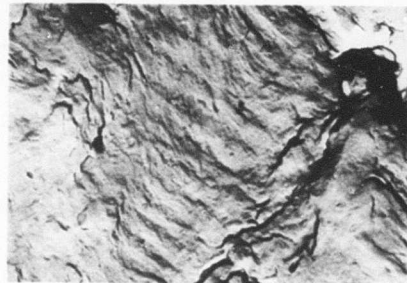


Magnification: 2,500×          Magnification: 10,000×

EX-78784, Serial Number CX 9069

Figure 44.   Fractographs of Surface of Failure of Gear Tooth Number 2 Showing
Failure Contour Typical of Fatigue.

Magnification: 2,500X                Magnification: 10,000X

EX-78784, Serial Number CX 9069

Figure 45.  Fractographs of Surface of Failure of Gear Tooth Number 3
Showing Failure Topography Typical of Fatigue.
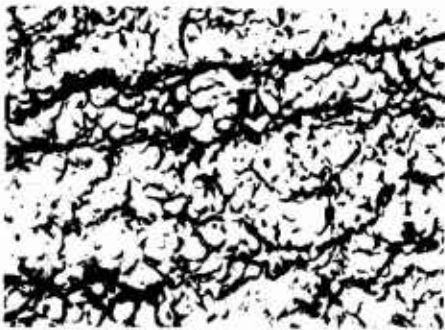


Magnification: 2,500X                Magnification: 10,000X

EX-78784, Serial Number CX 9069

Figure 46.  Fractographs of Surface of Failure of Gear Tooth Number 4
Showing Failure Topography Typical of Fatigue.

Magnification: 100X
Etchant: Vilella's Reagent
EX-78773, Serial Number CX 9077
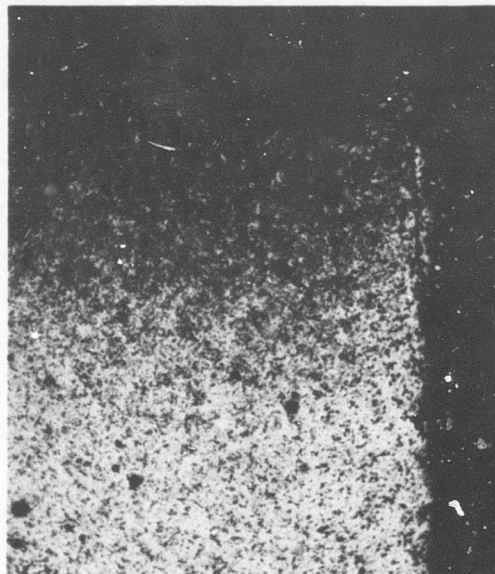
Figure 47. Photomicrograph of Transverse Section Through Failure Surface of Failed
Tooth Showing Straight-Line Failure Typical of Fatigue Originating in the Carburized
Case Hardened Root Radius.
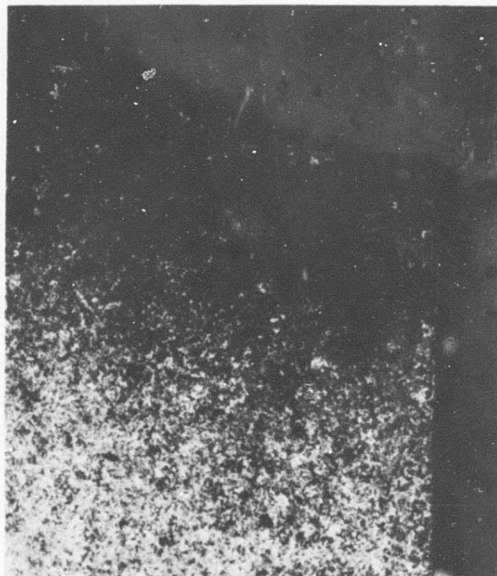


Magnification: 100X
Etchant: Villella's Reagent
EX-78775, Serial Number CX 9100

Figure 48. Photomicrograph of Transverse Section Through Failure Surface of Failed
Tooth Showing Straight-Line Failure Surface Typical of Fatigue Originating in the Case
Hardened Root Radius.

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Magnification: 100×
Etchant: Vilella's Reagent
EX-78777, Serial Number CX 9059
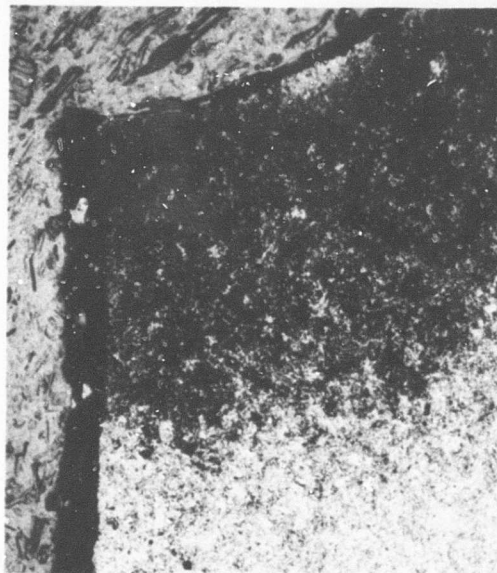
Figure 49. Photomicrograph of Transverse Section Through Failure Surface of Failed Tooth Showing Straight-Line Failure Surface Typical of Fatigue Originating in Carburized Case in the Root Radius.



Magnification: 100×
Etchant: Vilella's Reagent
EX-78779, Serial Number CX 7104

Figure 50. Photomicrograph of Transverse Section Through Failure Surface of Failed Tooth Showing Straight-Line Failure Surface Typical of Fatigue Originating in the Case Hardened Root Radius.

Magnification: 100×
Etchant: Villella's Reagent
EX-78782, Serial Number CX 9113
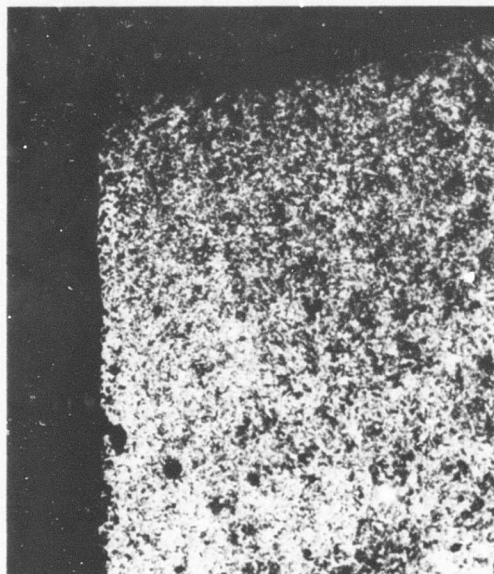
Figure 51.  Photomicrograph of Transverse Section Through Failed Tooth Showing Straight-Line Failure Typical of Fatigue Through a Carburized Case on Martensitic Microstructure.
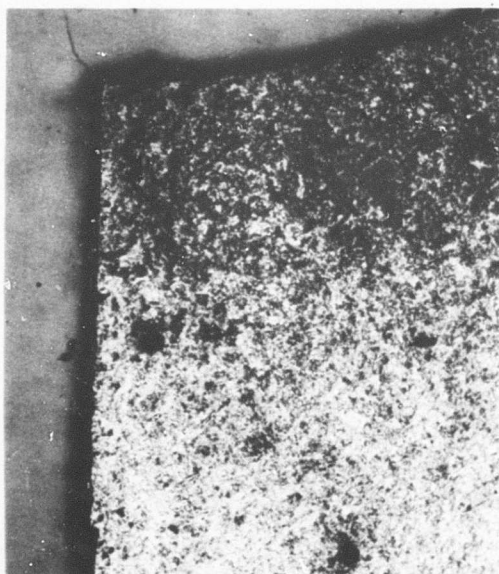


Magnification: 100×
Etchant: Vilella's Reagent
EX-78784, Serial Number CX 9069

Figure 52.  Photomicrograph of Transverse Section Through Failure Surface of Failed Tooth Showing a Straight-Line Failure Surface Typical of Fatigue Through Case Hardened Microstructure.
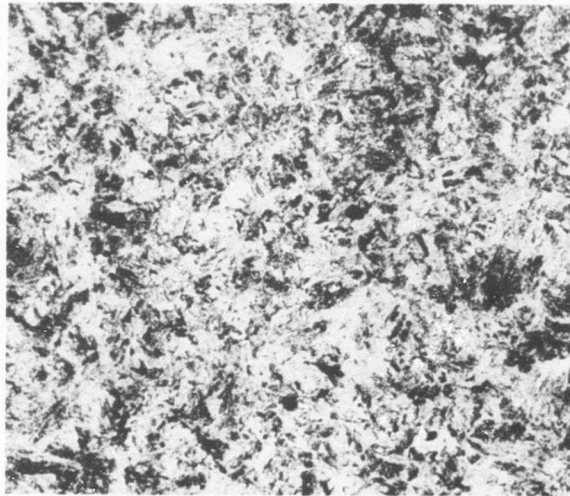
Magnification: 250X
Etchant: Vilella's Reagent
EX-78777, Serial Number CX 9059

Figure 53.   Photomicrograph of Transverse Section Through Test Gear Showing
Typical Core Structure of Tempered Martensite.
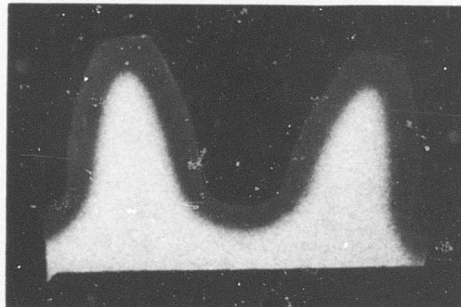


Figure 54.   Photograph of Section Through Test Gear Showing
Case Depth Around Root Fillet Contour.
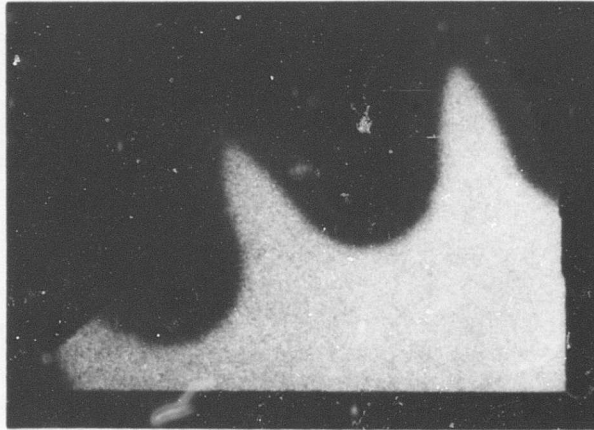
Magnification: 6X                    EX-78775, Serial Number CX 9100

Figure 55.   Photograph of Section Through Test Gear Showing
Case Depth Around Root Fillet Contour.
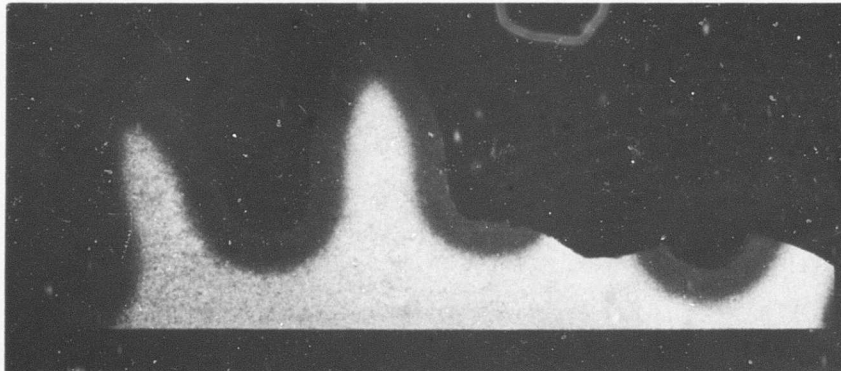


Magnification: 6X                    EX-78777, Serial Number CX 9059

Figure 56.   Photograph of Section Through Test Gear Showing Case
Depth Around Root Fillet Contour.
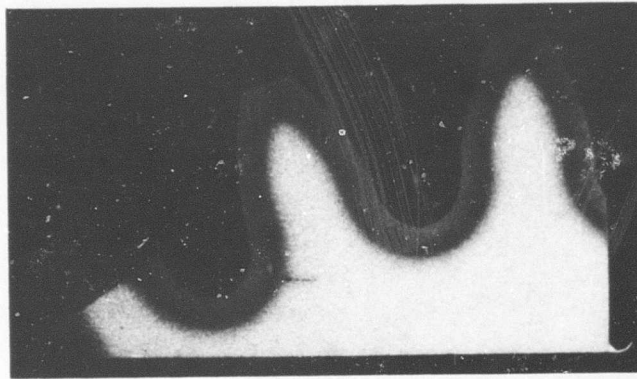
Magnification:  6X          EX-78779, Serial Number CX 9104

Figure 57.  Photograph of Section Through Test Gear Showing
Carburized Case Depth Around Root Fillet Contour.
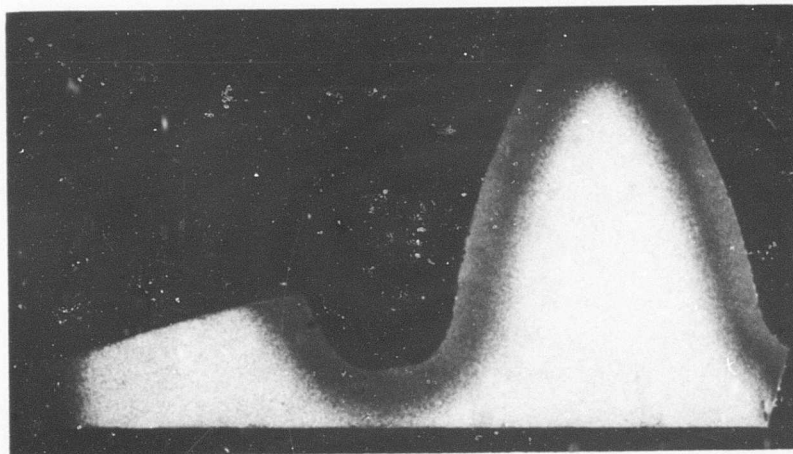


Magnification:  6X                    EX-78782, Serial Number CX 9113

Figure 58.  Photograph of Section Through Test Gear Showing Carburized
Case Depth Around Root Fillet Contour.
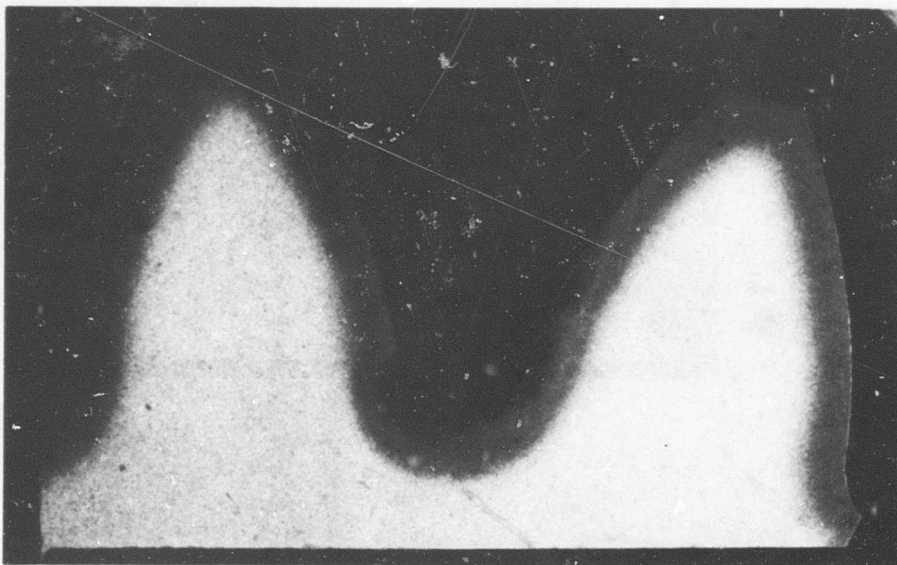
Magnification: 6X                        EX-78784, Serial Number CX 9069

Figure 59.  Photograph of Section Through Test Gear Showing Carburized
Case Depth Around Root Fillet Contour.
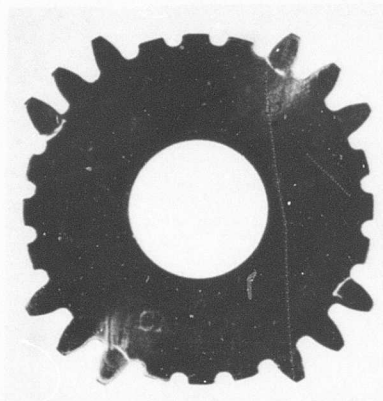


Magnification: 1X
EX-78775, Serial Number CX 9077

Figure 60.  Blacklight Photograph of Test
Gear Showing Cracks Indicated by
Fluorescent Penetrant Inspection
in Root Radii of Tested Teeth.
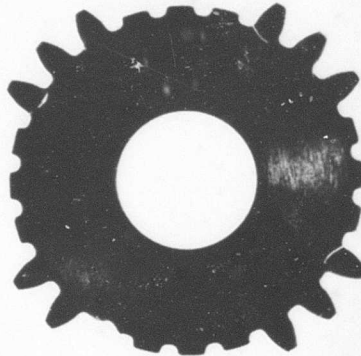
Magnification: 1X
EX-78775, Serial Number CX 9100

Figure 61.   Blacklight Photograph of Test Gear Showing
Cracks Indicated by Fluorescent Penetrant Inspection in
Root Radii of Failed Teeth.
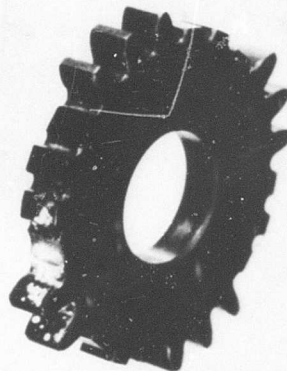


Magnification: 1X
EX-78777, Serial Number CX 9059

Figure 62.   Blacklight Photograph of Test Gear Showing
Cracks Indicated by Fluorescent Penetrant Inspection in
Center Root Radius Adjacent to Failed Tooth.
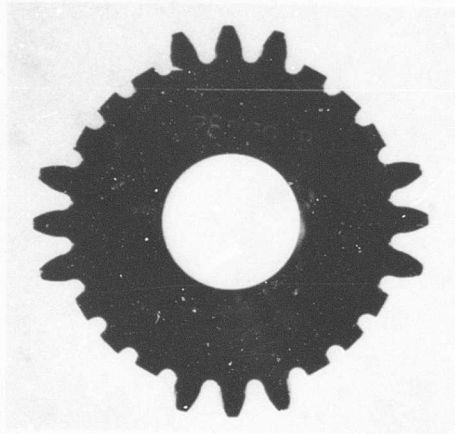
Magnification: 1X
EX-78779, Serial Number CX 9104

Figure 63.  Blacklight Photograph of Test Gear Showing Cracks
Indicated by Fluorescent Penetrant Inspection in Root
Radii of Failed Teeth.
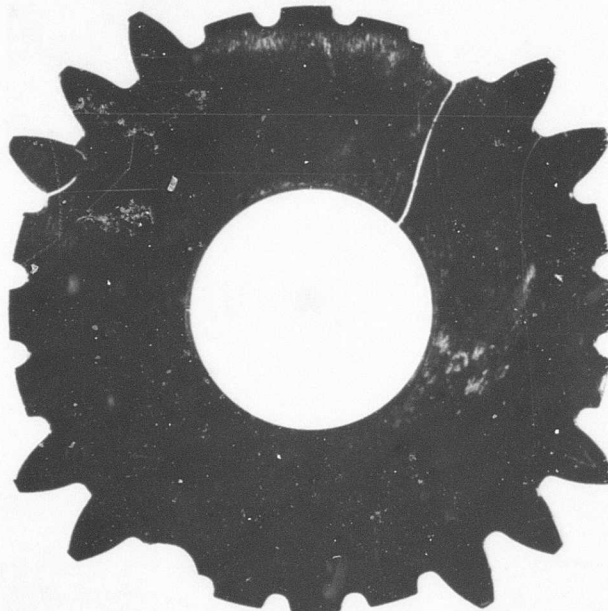


Magnification: 1X
EX-78782,
Serial Number CX 9113

Figure 64.  Blacklight Photograph of Test Gear Showing Radial Crack and Failed Teeth.
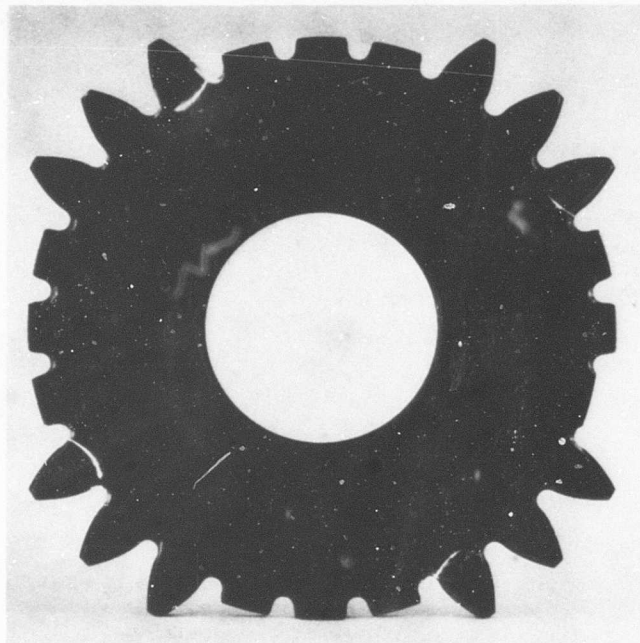
Figure 65. Blacklight Photograph of Test Gear Showing Cracks Indicated by Fluorescent Penetrant Inspection in Root Radii of Teeth 1, 2, 3, and 4.
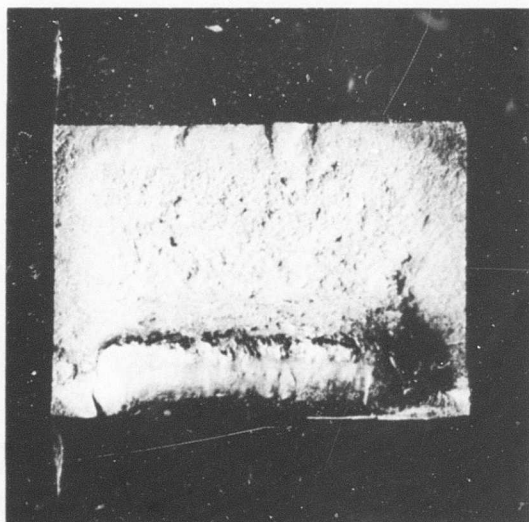
Figure 66. Photomicrograph of Surface of Failure of Tooth From Test Gear.

Magnification: 9X
EX-78775, Serial Number CX 9100
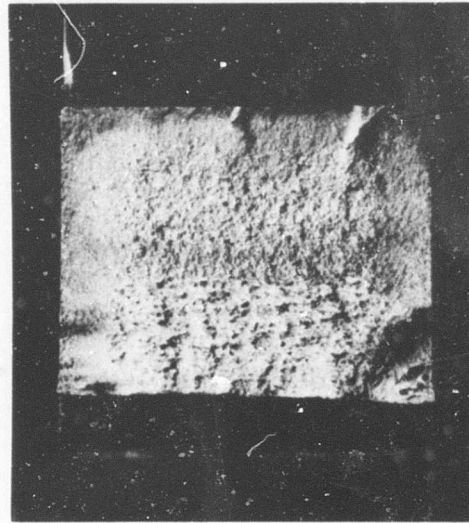
Figure 67. Photomicrograph of Surface of Failure of Failed Tooth From Test
Gear Showing Flat Failure in Root Radii of Teeth.
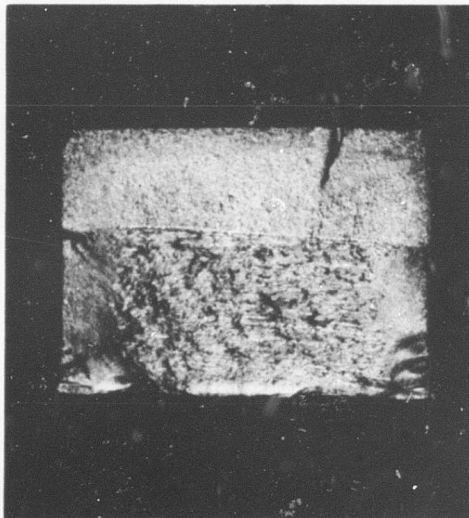


Magnification: 9X
EX-78779, Serial Number CX 9104

Figure 68. Photomicrograph of Surface of Failure of Tooth From Test Gear.
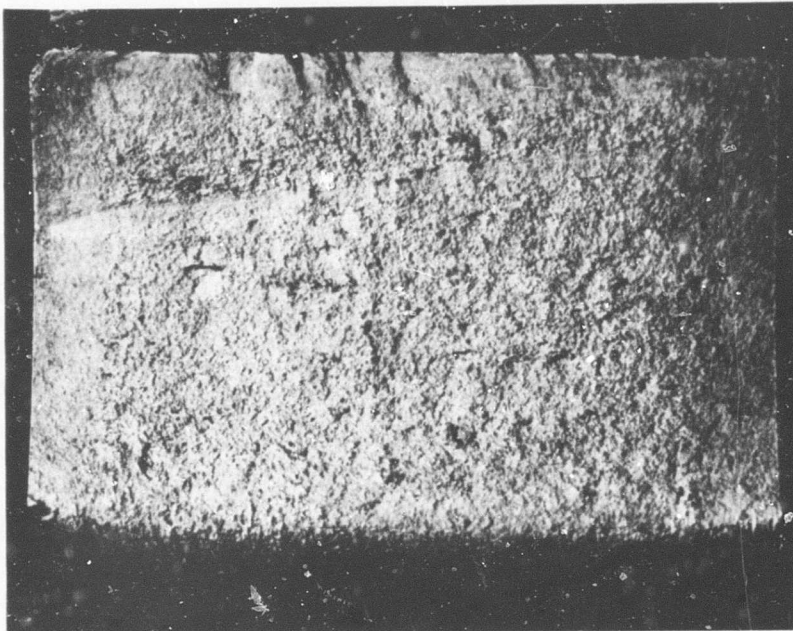
Figure 69.   Photomicrograph of Surface of Failure of Tooth 1 of Test Gear Showing
Multiple Origins of Failure in Root of Loaded Involute – No Typical Arrest Lines of
Fatigue Progression.
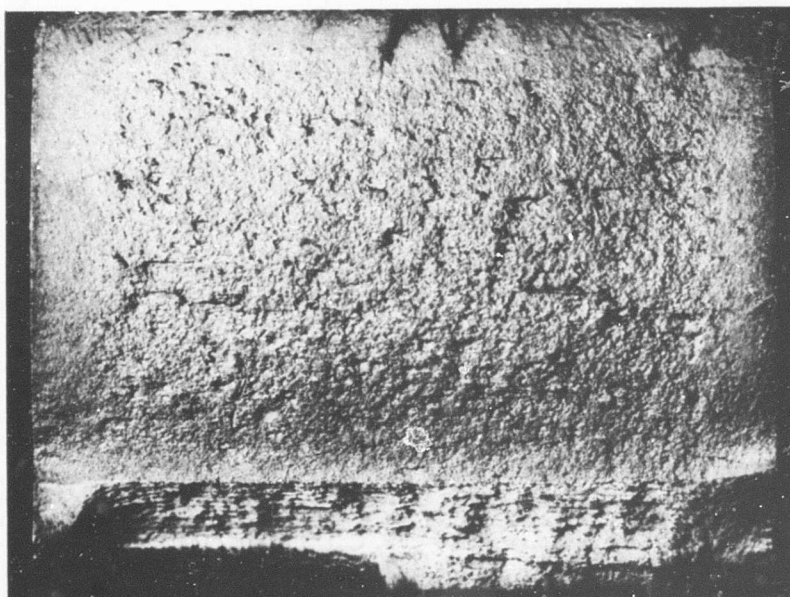
Figure 70.   Photomicrograph of Surface of Failure of Tooth 3 of Test Gear Showing
Multiple Origins of Failure and No Distinct Arrest Lines Typical of Fatigue Progression.

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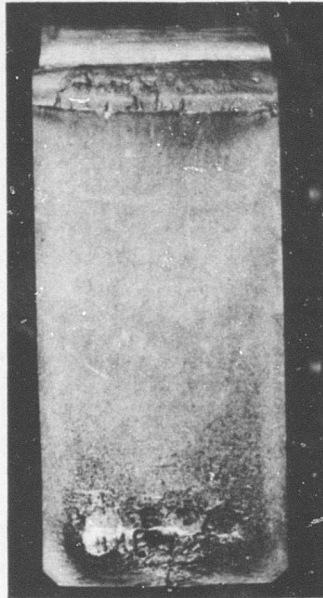
Magnification: 5X
EX-78782
Serial Number CX 9113

Figure 71.  Photomicrograph of Radial Surface of Failure of Test Gear Showing
Marks of Fatigue Progression From Below the Root to the Hub.

TABLE XII
RECORD OF HARDNESS GRADIENT TESTS OF TEST GEARS

| Depth Below Carburized Surface (inch) | $R_c$ Readings | | | | | |
|---|---|---|---|---|---|---|
| | EX-78773 CX 9077 | EX-78775 CX 9100 | EX-78777 CX 9059 | EX-78779 CX 9104 | EX-78782 CX 9113 | EX-78784 CX 9069 |
| 0.002 | 61 | 62 | 61 | 62 | 61 | 61 |
| 0.005 | 61 | 61 | 60 | 61 | 61 | 61 |
| 0.010 | 60 | 60 | 58 | 59 | 60 | 62 |
| 0.015 | 56 | 58 | 57 | 55 | 57 | 62 |
| 0.020 | 55 | 58 | 57 | 54 | 57 | 57 |
| 0.025 | 55 | 54 | 55 | 54 | 55 | 57 |
| 0.030 | 51* | 51* | 53 | 55 | 53 | 56 |
| 0.035 | 46 | 46 | 51 | 55 | 51 | 56 |
| 0.040 | 42 | 46 | 51* | 53* | 48* | 55 |
| 0.045 | 40 | 44 | 47 | 47 | 46 | 52 |
| 0.050 | 42 | 45 | 46 | 48 | 46 | 52* |
| 0.055 | 42 | 43 | 45 | 46 | 44 | 48 |
| 0.060 | 41 | 43 | 45 | 45 | 43 | 45 |
| 0.065 | 41 | 41 | 42 | 44 | 44 | 46 |
| 0.070 | 41 | 41 | 42 | 43 | 43 | 45 |
| 0.075 | — | — | — | 42 | 42 | 45 |
| 0.080 | — | — | — | — | 42 | 45 |
| 0.085 | — | — | — | — | — | 43 |
| 0.090 | — | — | — | — | — | 43 |

* Approximate effective case depth.
All hardness readings were taken at the root radii adjacent to the failure surface.

## R. R. MOORE TESTS

**R. R. Moore** test specimens **were** manufactured from the same heat of material as the **test gears.** Manufacturing **followed** heat treating and grinding routings used for the **gears as closely as feasible. The** process routing for the specimens is presented in **Table XIII. The test results are** given in Table XIV.

### TABLE XIII
### SPECIMEN PROCESS ROUTING PROCEDURE

1. Carburize and anneal per EPS* 202 to an effective case depth of 0.035 inch as determined by the fracture specimen.
2. Harden and temper per EPS 202 and PCI** 8000 and stabilize per EPS 202.

   Core Hardness — $R_c$ 40
   Case Hardness — $R_{15/N}$ 90 ($R_c$ 60)

3. Grit blast with 80-grit shot.
4. Remove 0.010 to 0.016 inch from outside diameter by grinding.
5. Stress relieve per EPS 202 and PCI 8000.
6. Nital etch per EIS † 1510.
7. Shot peen per EPS 12140 followed by EPS 12176.
8. Stress relieve per EPS 202 and PCI 8000.
9. Coat with black oxide per AMS-2485.

   * Allison Engineering Processing Specification.
   ** Allison Process Control Instruction.
   † Allison Engineering Inspection Specification.

## EXPERIMENTAL INVESTIGATIONS

In this phase of the program, photostress and strain gage measurements were used to investigate the location and magnitude of the maximum bending stress.

By cementing a sheet of special plastic* to the gear face (actual fatigue test gear) and trimming to the contour of the test tooth, it was possible to obtain indications of stress distribution, stress values along the tooth contour, and maximum stress locations. A large field reflection polariscope (LF/Z meter) and a telemicroscope were used to study in some detail the point of high stress.

To complement the photostress analysis, strain gages were installed in the root of the gear tooth at the theoretical point of maximum stress as shown in Figure 72. The gear was mounted to the fatigue test rig and loaded by means of the bias spring.

The protuberance hobbed gear, part number EX-78776 (with a 20-degree pressure angle and a minimum fillet radius), was selected for stress analysis.

The plastic sheet manufacturer supplied the calibration of the optical strain constant of 1080 microinches per inch per fringe or tint-of-passage (sharp line between red and blue).

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*Special birefringent material, plastic sheet type S, 0.120 inch thick, Model Number X-10062, Instruments Division of The Budd Company, Phoenixville, Pennsylvania

TABLE XIV
R. R. MOORE TEST RESULTS

| Specimen Number | Stress (p.s.i.) | Test Cycles* (× 10³) | Surface Finish (microinches) | Failure Origin | Failure Location |
|---|---|---|---|---|---|
| 18 | 130,000 | 106,584 | 23 to 27 | Terminated | — |
| 17 | 135,000 | 105,951 | 25 to 28 | Terminated | — |
| 2 | 140,000 | 101,234 | 30 to 35 | Terminated | — |
| 6 | 140,000 | 102,384 | 25 to 30 | Terminated | — |
| 15 | 140,000 | 111,435 | 20 to 25 | Terminated | — |
| 14 | 150,000 | 74 | 25 to 30 | Surface | Off center *** |
| 1 | 150,000 | 138 | 32 to 37 | Surface | Slightly off center † |
| 4 | 150,000 | 50,683 | 30 to 35 | Subsurface ** | Center ‡ |
| 13 | 150,000 | 90,852 | 28 to 32 | Surface | Slightly off center |
| 11 | 150,000 | 103,034 | 8 to 13 | Terminated | — |
| 10 | 160,000 | 44 | 25 to 28 | Surface | Center |
| 7 | 160,000 | 134 | 12 to 20 | Surface | Off center |
| 5 | 160,000 | 3,317 | 25 to 30 | Surface | Center |
| 3 | 160,000 | 6,061 | 30 to 35 | Surface | Center |
| 16 | 170,000 | 74 | 25 to 30 | Surface | Slightly off center |
| 9 | 170,000 | 114 | 20 to 25 | Surface | Center |
| 8 | 170,000 | 187 | 10 to 15 | Surface | Center |
| 12 | 170,000 | 228 | 28 to 32 | Surface | Center |

\*    Arithmetic average.
\*\*   Within effective case.
‡    Center is midpoint of specimen.
†    Slightly off center is 1/16 to 1/4 inch from midpoint.
\*\*\*  Off center is 1/4 to 1/2 inch from midpoint.

The photostress gear was statically loaded in 1000-pound increments. Readings were taken at each 1000-pound step, and photographs were taken at zero and 4000 pounds. This load limit was chosen as the stopping point because the concentration of strain was so confined and was beyond the reading capacity of the LF/Z meter.

The greatest stress concentration, as measured by the LF/Z meter, occurred at the calculated point for the placement of the strain gages. The strain rate was 1080 microinches per inch (32,400 p.s.i.) per 1000 pounds of load by photostress and 1140 microinches per inch (34,200 p.s.i.) by strain gage. Figure 73 illustrates the stress distribution for the 4000-pound load point. Since monochromatic light was not used, both isoclinic lines (lines of stress direction) and tints-of-passage are seen as the darker lines and cannot be defined without the aid of the color photographs.
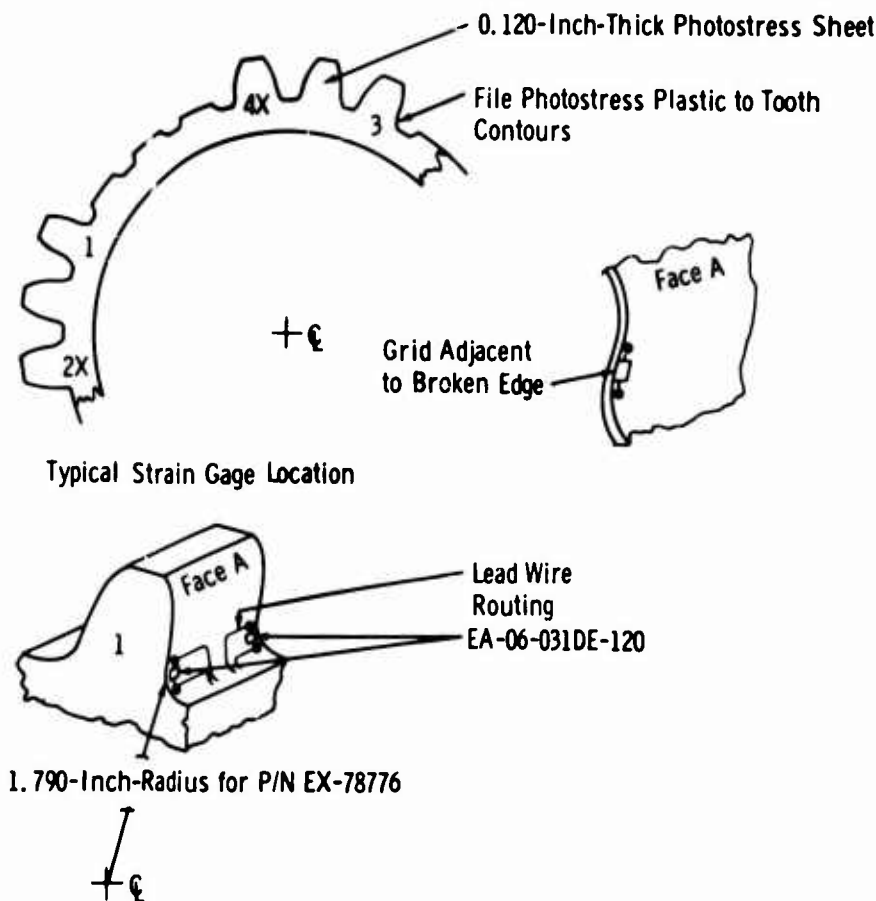
Figure 72. Schematic of Instrumentation on Photostress Gear.

To permit comparison of calculated stresses with actual measured stresses, one tooth from each of the eight 4-inch-pitch-diameter gears was instrumented with strain gages. Static strain versus load at the high point of single tooth contact was obtained. Each gear was instrumented with strain gages as shown in Figure 74. The radial location of the gages was at the expected crack point based on crack measurements from the gears (diametral pitch = 12) that were available at the time.

The gears were tested on the fatigue test rig using the same procedure for installation as used for fatigue and photostress tests. The results of the data are shown in Figures 75 and 76. The gages were located on the tension side except for one on the compression side of one gear.

DYNAMIC TESTS

The effect of speed on bending stress can be categorized as follows.

- Centrifugal stress, a steady-state stress at any particular speed caused by internal forces. As noted in Figure 77, this effect consists of tensile stresses in the tooth and hoop stresses in the gear rim.
- Dynamic stress, a cyclic stress with a constant peak magnitude at any particular speed caused by tooth load, imperfect tooth meshing, load sharing, and other geometrical and manufacturing properties of the gear. It is cyclic since it occurs only when the tooth is under load, e.g., in mesh with a mating gear. This is shown graphically in Figure 78.

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**P/N EX-78776**
**(20-Degree Pressure Angle)**

First Tint — Second Tint
Second Tint
Load Point
Location of Maximum Stress
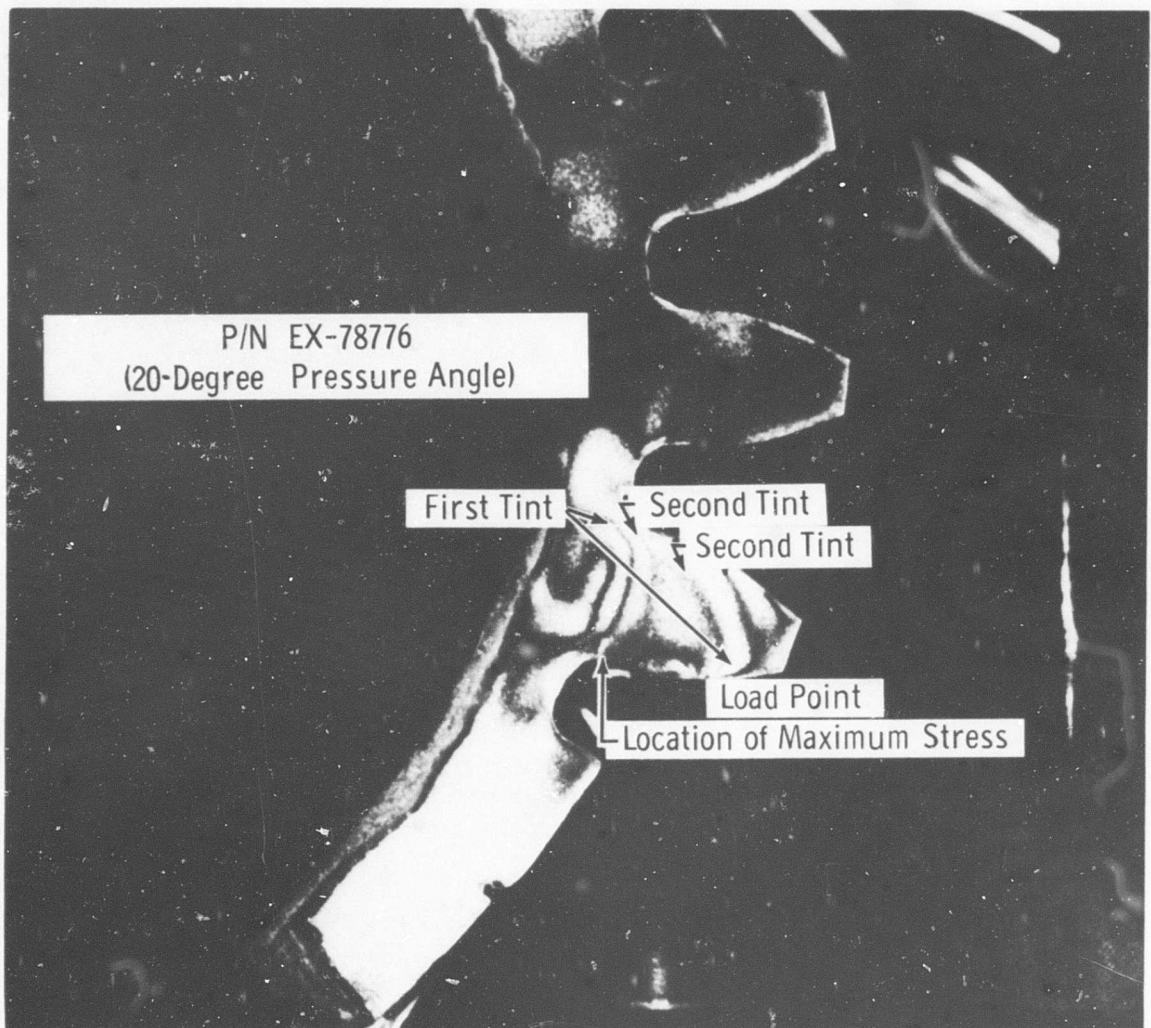
Figure 73. Gear Tooth Showing Photostress Pattern at 4000-Pound Load.

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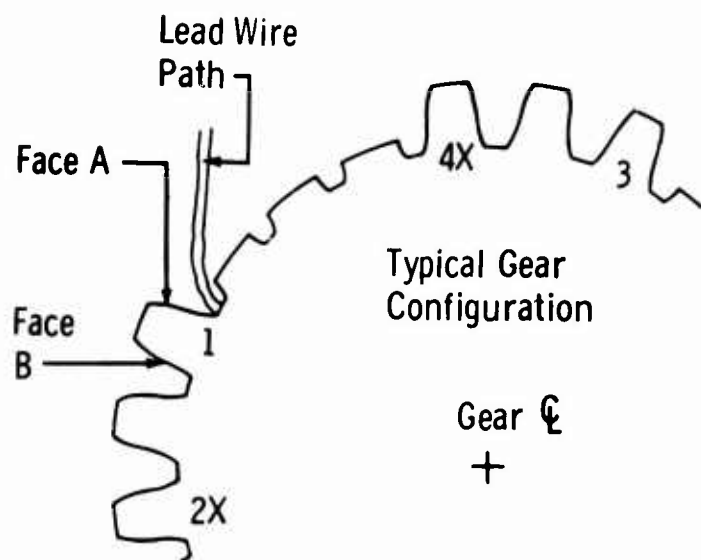
**Strain Gage Mounting Procedure**

1. Vapor blast to remove black oxide.
2. Wipe with W. T. Bean neutralizer.
3. Attach strain gage with Eastman No. 910 contact cement.
4. Protect gage with Dow Corning silicon wax fluid F145.
5. Attach 4-foot-long lead wires.

EA-06-031DE-120—
Two Required per Tooth
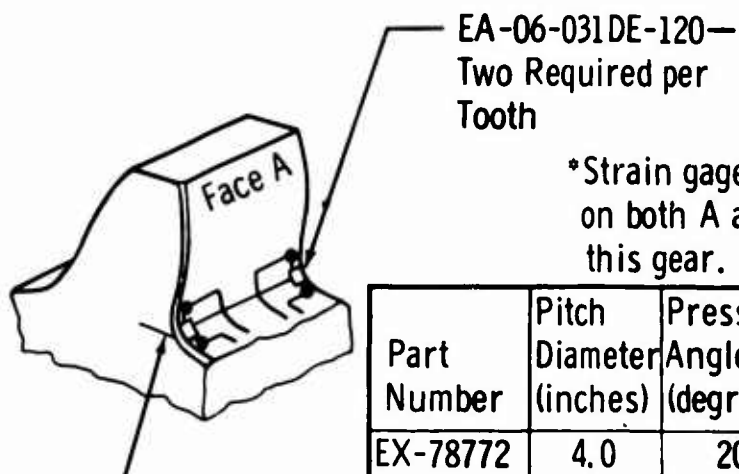
*Strain gages to be installed on both A and B faces on this gear.

| Part Number | Pitch Diameter (inches) | Pressure Angle (degrees) | Serial Number | Tooth Number | Radius, R |
|-------------|-------------------------|--------------------------|---------------|--------------|-----------|
| EX-78772 | 4.0 | 20 | CX9090 | 4 | 1.7959 |
| EX-78774 | 4.0 | 20 | CX9066 | 1 | 1.8023 |
| EX-78776 | 4.0 | 20 | CX9007 | 2 | 1.7713 |
| EX-78778 | 4.0 | 20 | CX9056 | 3 | 1.7781 |
| | | | | | |
| EX-78780* | 4.0 | 25 | CX9096 | 4 | 1.7804 |
| EX-78782 | 4.0 | 25 | CX9111 | 4 | 1.8058 |
| EX-78784 | 4.0 | 25 | CX9071 | 4 | 1.7741 |
| EX-78786 | 4.0 | 25 | CX9012 | 1 | 1.7751 |

Lay out scribe marks as shown on both sides. Then draw line between scribe marks. Locate strain gage grid on scribe line adjacent to edge break on face A.

**Figure 74.** Schematic of Strain Gage Instrumentation for 4-Inch-Pitch-Diameter Gear.
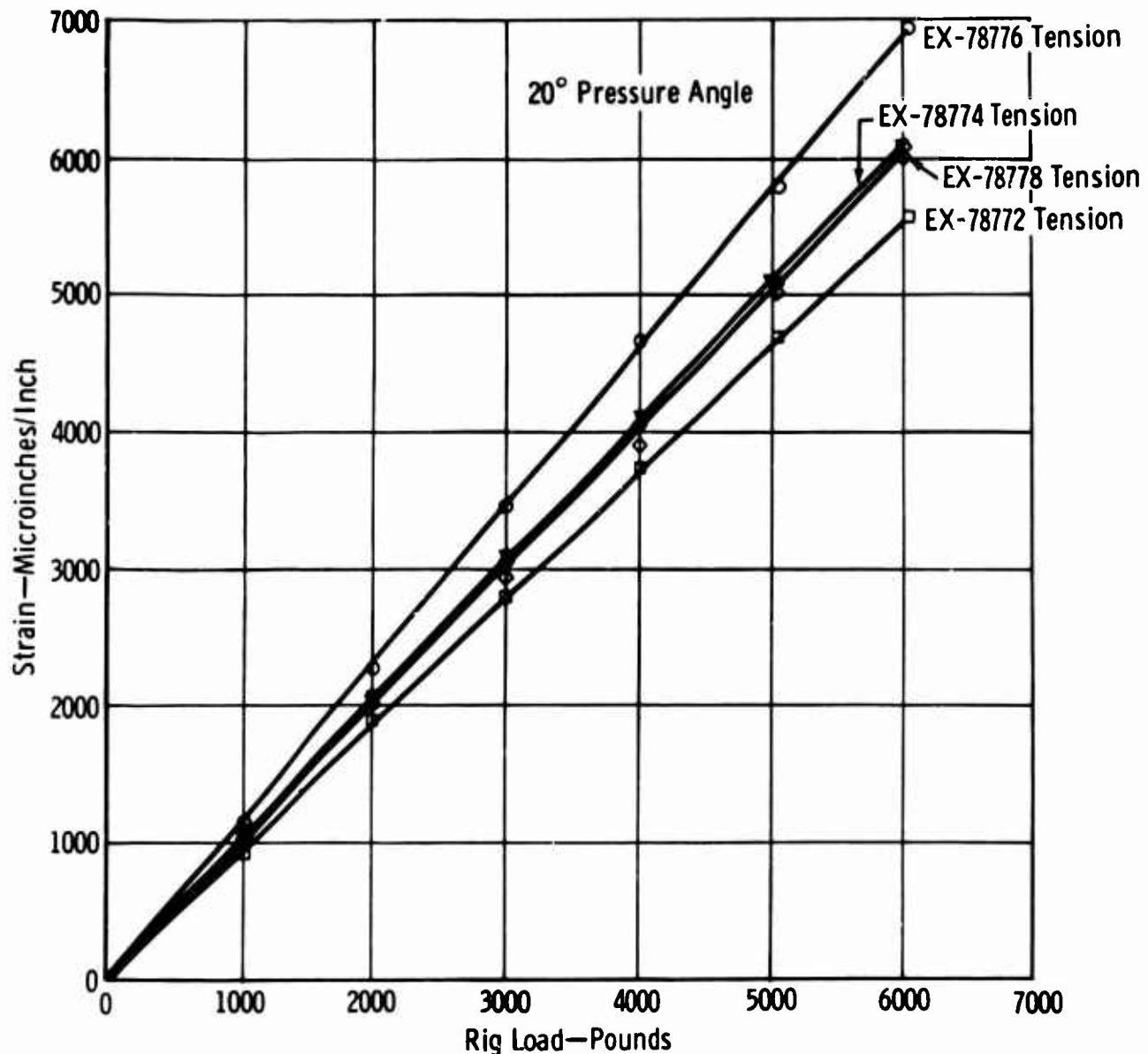
Figure 75.   Calibration Curve for Gear Test Rig – 20-Degree Pressure Angle.

As shown in Figure 78, doubling the speed not only increases the frequency of the dynamic stress, but also raises the centrifugal stress level and the amplitude of the dynamic stress.

To better understand the effects of speed on gear tooth bending stress, a gear was instrumented and strain data were recorded during actual running conditions. Data were recorded to 26,500 feet per minute pitch-line velocity. The gear tested was the propeller brake outer member (part number 6829395) in a 501-D13 turboprop engine gearbox. The instrumentation consisted of strain gages located on the tooth as shown in Figure 79. One tooth had gages located on the tension side and another tooth, 180 degrees, had gages on the compression side. Two gages were located in the root and two at the point of expected maximum stress in the root fillet.
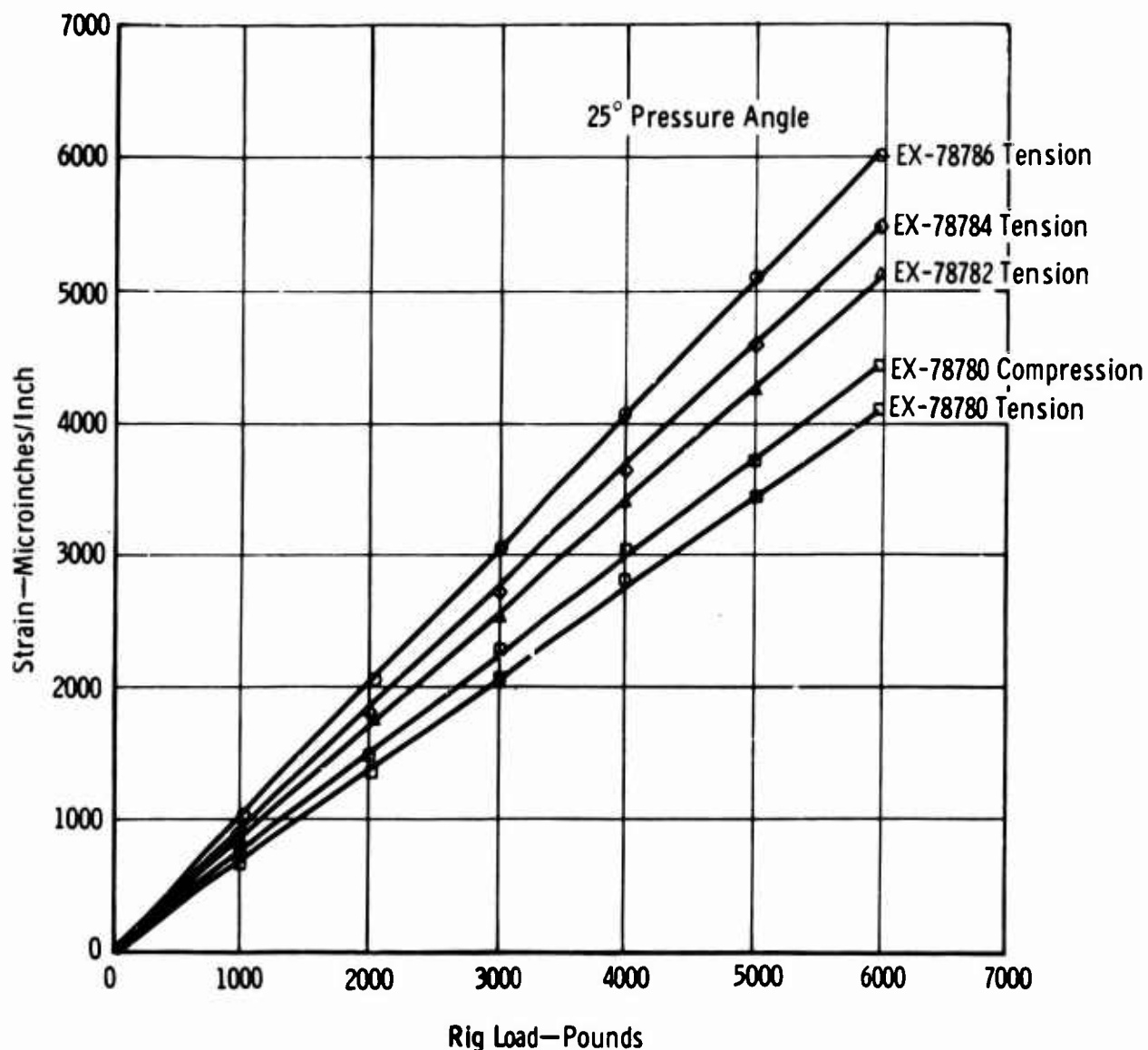
Figure 76. Calibration Curve for Gear Test Rig— 25-Degree Pressure Angle.

By means of electronic test data recording, the centrifugal stress and the dynamic stress were separated. This was possible since centrifugal stress is a steady-state stress and dynamic stress is a cyclic stress. The centrifugal stress was obtained by taking strain gage readings under zero-load conditions at various speeds. The dynamic stress was taken under loaded conditions and was the peak strain reading above the centrifugal base line.

The gear train used is shown schematically in Figure 80. The power input was through the main accessory drive gear which mated with the test gear. The load was applied by means of a water brake attached to the alternator drive. To calibrate the strain gages, torque was applied in a static condition. The instrumented tooth was rolled through the highest load point for maximum stress calibration. This setup is shown in Figure 81. The test gear and mating gear meet AGMA class 10 to 12 tolerances. The gear geometry and tolerances are shown in Figure 82.

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**Centrifugal**
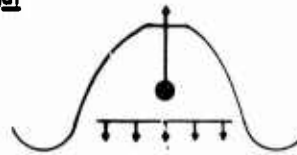
Radial Tensile Stress
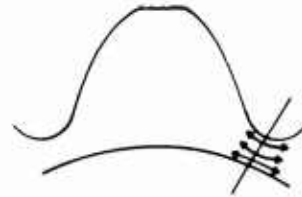
Hoop Stress (Circumferential Tensile)

**Dynamic**

Tangential Tooth Load

Cyclic Bending Stress

AGMA Stress
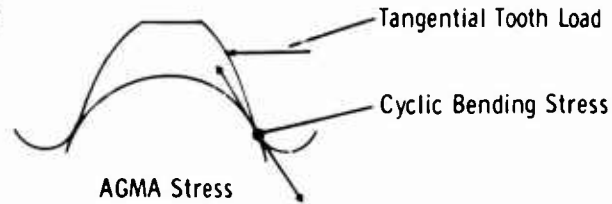
Figure 77.   Gear Tooth Bending Stress Schematic.



Speed = 2N

Stress

Centrifugal Stress

Peak Dynamic Stress

1 Revolution

Speed = N

Time
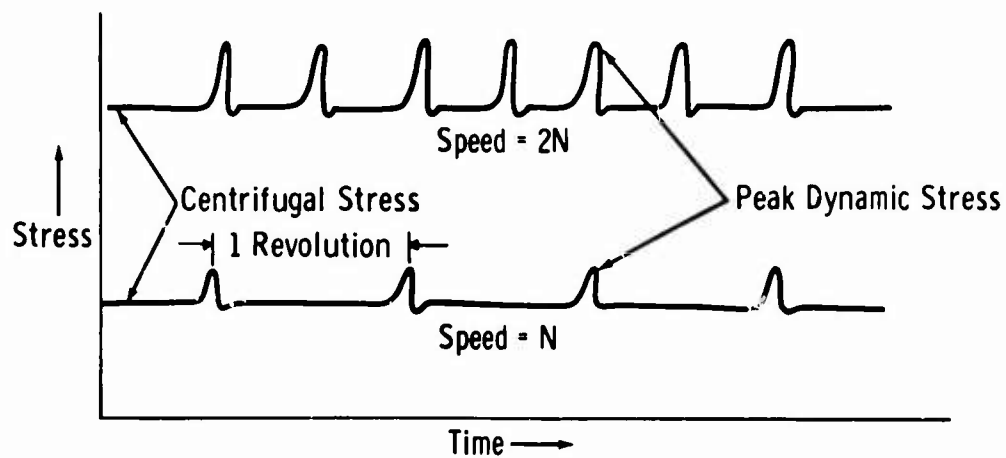
Figure 78.   Diagram Showing Effect of Speed on Gear Tooth Stresses.
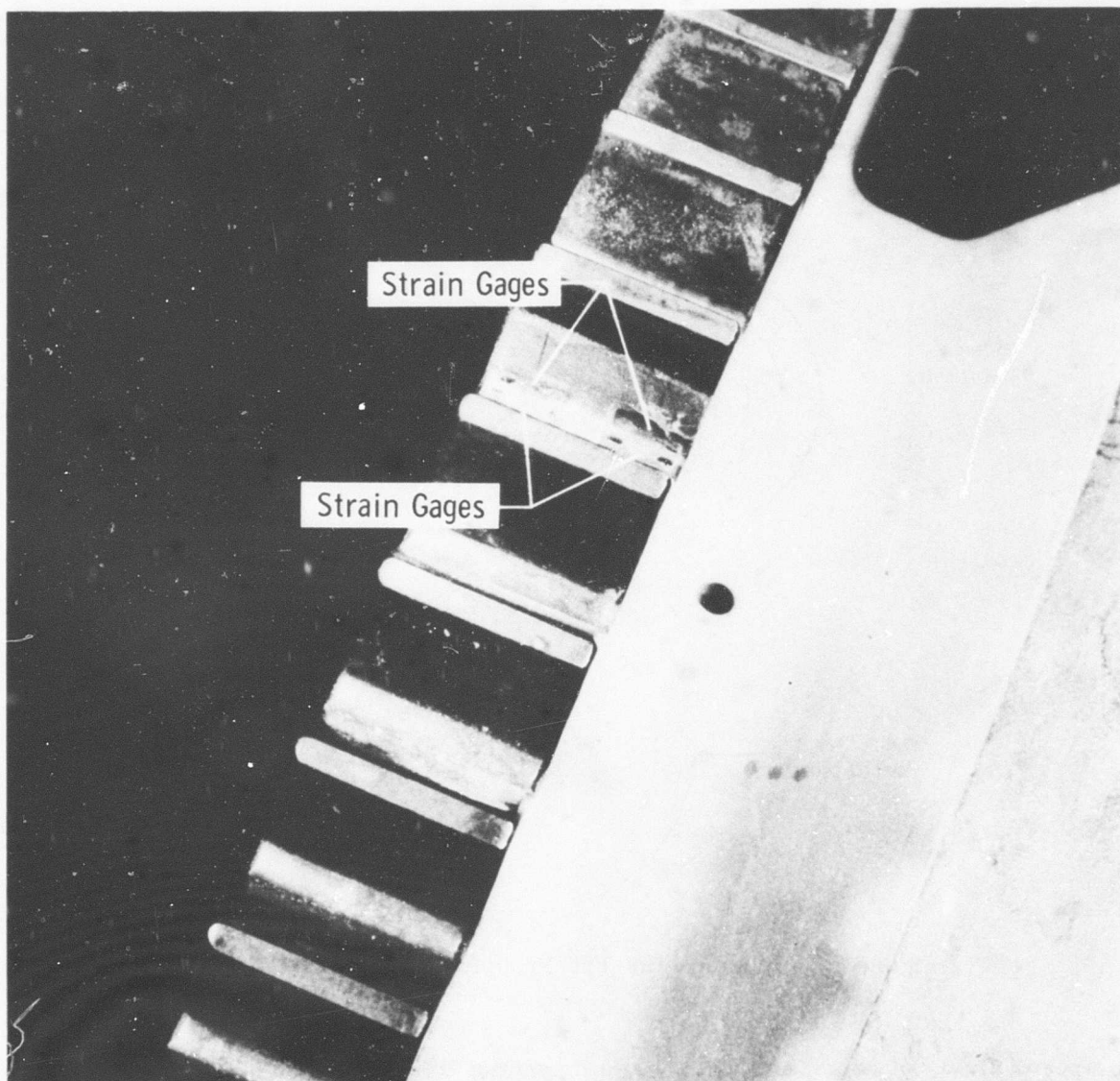
Figure 79. Dynamic Test Gear Strain Gage Instrumentation.

To isolate the stresses due to speed effects in the tooth root, the instrumented gear was first tested at zero load in the reduction gearbox. Using a three-wire strain gage hookup and allowing gearbox oil temperatures to stabilize, strain due to centrifugal loads was recorded. Testing was conducted at essentially zero tangential loads for speeds varying from 10,000 to 15,000 r.p.m. Figure 83 shows the centrifugal strain (tension) on the gear tooth.

The gear was then loaded by means of a water brake to obtain stress versus speed data. The strain gage instrumentation was routed through a slip-ring assembly, and the gage signal was recorded by a 16-channel Miller oscilloscope recorder. The gear was tested
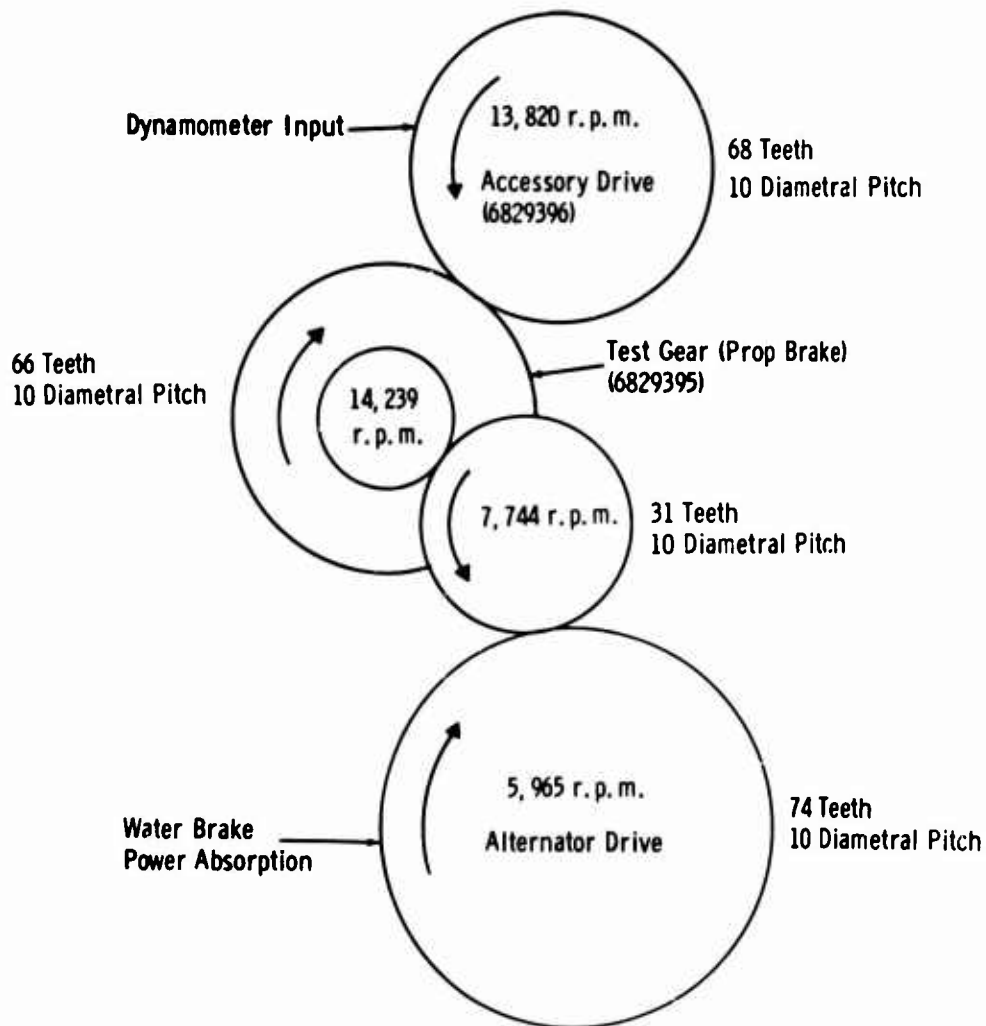
**Figure 80. Schematic of T56 Propeller Brake Gear Train.**

at speeds of 10,000 to 15,530 r.p.m. and tangential loads of 350 to 950 pounds. Figure 84 shows data from foui strain gages. The data shown represent the average strain range at the speed at which the gear was tested. Of the eight gages installed, only these four survived the testing schedule.
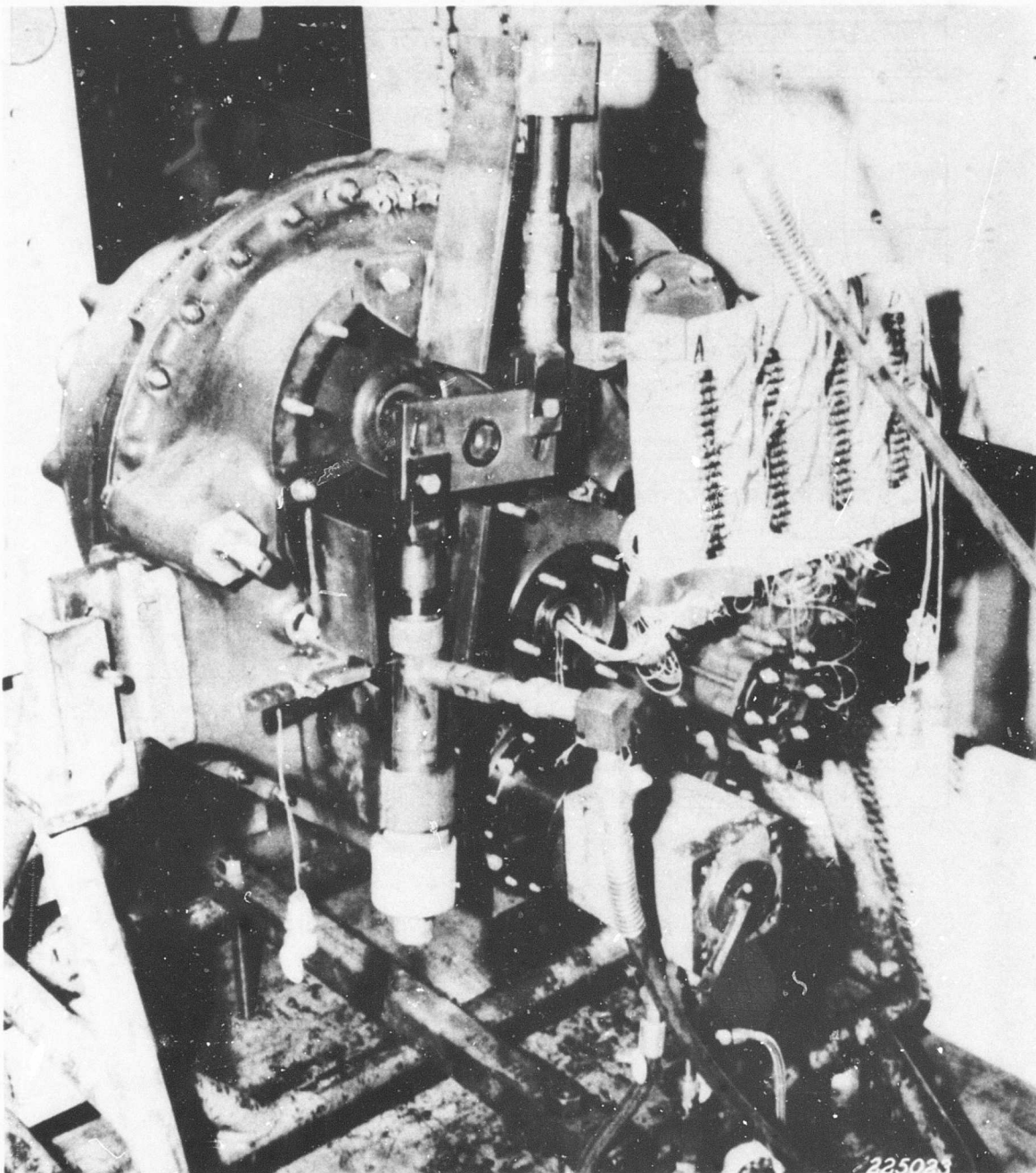
Figure 81.   T56 Gearbox Used for Dynamic Gear Test.

Propeller Brake Outer Gear 6829395
(dynamic test gear)

Accessory Drive Gear 6829396 (driving gear)

| GEAR TOOTH CONTROL | | | | |
|---|---|---|---|---|
| INVOLUTE PROFILE TOLERANCE | | | | |
| SIDE A (Driven) | | OD | SIDE B (Coast) | |
| 0.9° | -0.0001 | ODB | | +0.0003 |
| 3.3U | +0.0005 | | | -0.0007 |
| 4.3° | +0.0005 | PD | | +0.0006 |
| 15.1U | +0.0001 | | | -0.0000 |
| 7.6° | | MFCA | | |
| 27.0 U | | | | |
| 8.5° | | APD | | |
| 30.3 U | | | | |
| 31.0° | | BCD | | |
| 110.0U | | | | |

| SPACING TOLERANCE | | | |
|---|---|---|---|
| A 0.0005 | | B 0.0007 | |

| LEAD TOLERANCE | | |
|---|---|---|
| A 0.0005 RH | B 0.0007 RH | |
| 0.0005 LH | 0.0007 LH | |

| FULLNESS TOLERANCE | | |
|---|---|---|
| A 0.0006 | B 0.0007 | |
| 0.0002 | 0.0001 | |
| MAXIMUM HOLLOW IN FORM | 0.0002 | |

NOTE      U = I UNIT = 0.0147 IN.
Maximum fullness to occur within
0.7° (2.5U) of PD.

| GEAR TOOTH CONTROL | | | | |
|---|---|---|---|---|
| INVOLUTE PROFILE TOLERANCE | | | | |
| SIDE A (Drive) | | OD | SIDE B (Coast) | |
| 0.9° | +0.0005 | ODB | | +0.0007 |
| 3.3U | -0.0001 | | | -0.0003 |
| 4.1° | +0.0007 | PD | | +0.0008 |
| 15.2U | +0.0003 | | | +0.0002 |
| 7.4° | | MFCA | | |
| 27.0U | | | | |
| 8.3° | | APD | | |
| 30.3U | | | | |
| 30.9° | | BCD | | |
| 112.9U | | | | |

| SPACING TOLERANCE | | | |
|---|---|---|---|
| A 0.0005 | | B 0.0007 | |

| LEAD TOLERANCE | | |
|---|---|---|
| A 0.0005 RH | B 0.0007 RH | |
| 0.0005 LH | 0.0007 LH | |

| FULLNESS TOLERANCE | | |
|---|---|---|
| A 0.0006 | B 0.0007 | |
| 0.0002 | 0.0001 | |
| MAXIMUM HOLLOW IN FORM | 0.0002 | |

NOTE      U = I UNIT = 0.0147 IN.
Maximum fullness to occur within
0.7° (2.6U) of PD.

10 pitch
66 teeth
25° Pressure Angle

Distance over two 0.1728-dia pins

6.8370 $^{+0.0000}_{-0.0041}$

PD run-out, 0.002

Face width, 0.375
Arc tooth thickness at PD

0.1541 $^{+0.0000}_{-0.0020}$

Base circle diameter—5.9816

10 pitch
68 teeth
25° Pressure Angle

Distance over two 0.1728-dia pins

7.0370 $^{+0.0000}_{-0.0041}$

PD run-out, 0.002

Face width, 0.628
Arc tooth thickness at PD

0.1541 $^{+0.0000}_{-0.0020}$

Base circle diameter—6.1629

0.006 to 0.010 backlash with mating gear on STD centers

Figure 82.  Dynamic Test Gear and Driving Gear Geometry and Tolerances.

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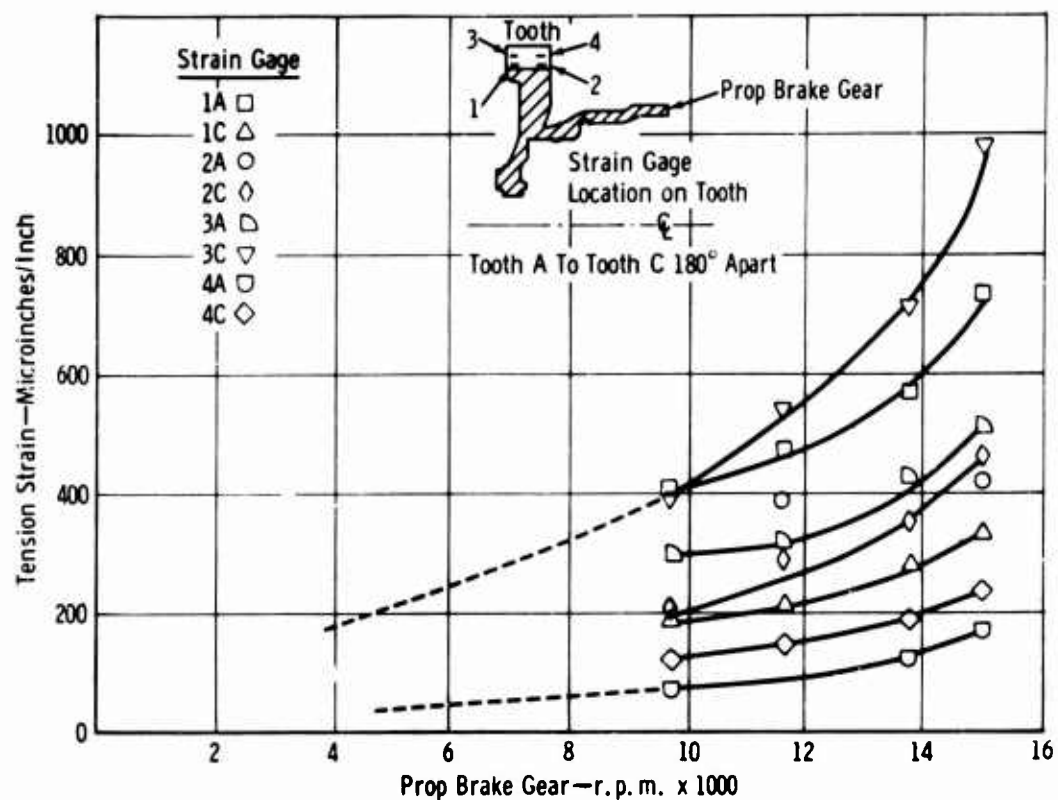
Figure 83. Effect of Speed on Gear Tooth at No-Load Condition.
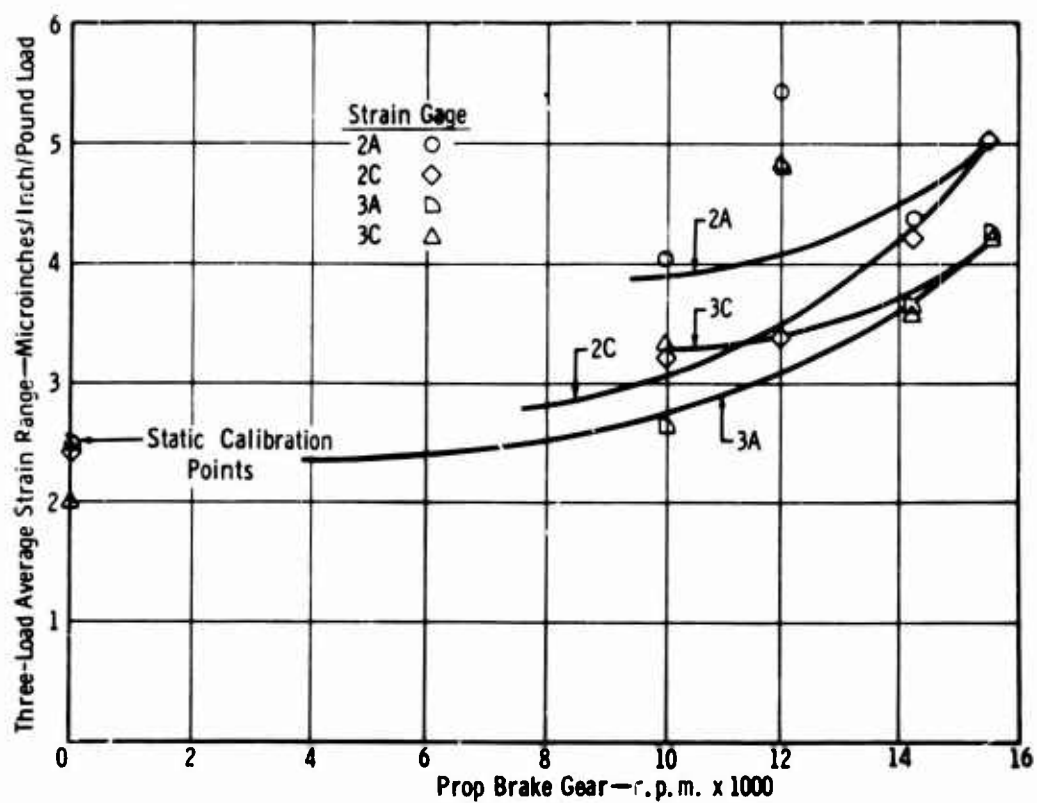


Figure 84. Effect of Speed on Loaded Gear Tooth.

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# DISCUSSION OF RESULTS

## EVALUATION PROCEDURE

**The test results** were evaluated by the following steps:

1. Determine predictive ability of the five calculation methods.

2. Compare strain gage and photostress data with calculated stress.

3. Determine significance of geometric variables based on most predictive calculation methods.

4. Determine basic material strength and design value.

5. Compare test data and design value to the literature.

6. Analyze centrifugal and dynamic load effects.

7. Establish computer program.

## PREDICTIVE ABILITY OF CALCULATION METHODS

The predictive ability of the five methods studied for calculating bending stress was evaluated by use of the mean endurance limits fitted through the fatigue test gear data points. Proportionality factors were used to convert the unit load endurance limits for each gear configuration to endurance limit values based on each of the stress calculation methods. These endurance limit values are listed in Table XV and are ranked in descending order. Average, range, and variation in endurance strength for each calculation method are also given. The AGMA method produced the smallest variation which is considered to be one of the best criteria for evaluation of the various calculation methods. Also, the test rig (applied load) ranked all the larger (6-diametral-pitch) gears first as would be expected. However, the Heywood and Kelley-Pedersen method also ranked all but one of the large gears first, indicating that these calculation methods may not adequately compensate for changes in diametral pitch.

Further analyses were made by comparing the rank given to each test gear configuration by each calculation method with the test rig load endurance limit ranking. Since a high stress should result in a low life, the calculated stress rankings were inverted. The results of this comparison are given in Table XVI. The AGMA formula predicted the greatest number of correct rank positions (6 out of 16) and also had the best average prediction accuracy (within 1.25 rank positions).

The endurance limit for fatigue test gear configuration number 3 appears to be abnormally low. See Table XV. It was therefore deleted from critical calculations (range and variation) but not from averages. This configuration (part number EX-78774) did have dimensional discrepancies (0.070-inch root fillet radius instead of 0.080-inch minimum print requirement). This should have lowered the life to approach that of configuration number 1, which is the same except for 0.050-inch minimum root fillet radius. The life was actually only two-thirds of that of configuration number 1. The test data had very low fatigue life scatter, which may be indicative of a severe stress concentration. Since the low endurance life was not determined until late in the program, no metallurgical investigations of this gear were accomplished.

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Continued analysis of the fatigue test results based on individual measured physical dimensions rather than part number drawing dimensions could appreciably increase the confidence level of the results. The test results of one gear have been corrected to a 10-percent lower stress level to adjust for a 0.010-inch oversize root diameter. Thus, correction of all data to compensate for individual sizes within the ± 0.002-inch root diameter drawing tolerance would adjust relative calculated stresses by approximately 4 percent. Similar changes could be made for individually measured tooth thicknesses and fillet radii. The protuberant hobbed configurations could be revised, based on measured hob dimensions.

To accomplish the individual analysis described for each fatigue test tooth would require conversion of the present computer program to permit operation on the smaller IBM 1130 rather than on the IBM 7094. The program would also require revision to eliminate unnecessary output and thus would avoid overloading the smaller computer. Also, the input would have to be modified to use the measured dimensions directly. Table XVII lists the critical root diameter, root fillet radius, and over-pin dimensions for each gear.

Each fatigue test gear tooth was examined to determine and record the edge break condition in the failure region. See Table XVII. These edge breaks were not as consistent as desired due to the difficulty of controlling a hand operation. Direct comparison of edge break and fatigue life failed to indicate any general influence of edge break on the test results.

STRAIN GAGE DATA

Evaluation of the static strain gage measurements confirmed the validity of the AGMA method of calculating bending strength. Table XVIII shows the measured strain gage data in terms of strain rate for each configuration tested. The remaining columns show a comparison of the various methods of calculating bending strength in terms of strain rate. The percent deviation shows the magnitude of difference between the measured and calculated strain for each configuration. The AGMA method produces a minimum difference for each configuration. The last column shows the stress concentration factor calculated from the difference between the Lewis calculated and the measured data.

To further indicate the degree of correlation, Figure 85 shows stress versus load for the measured data and the AGMA calculation. The percent deviation of the calculated stress from the measured stress is shown in Figure 86. The present AGMA method gave the smallest deviation from the measured stress.

Since none of the formulas considered fillet configuration, the data were split into two groups—full form ground and protuberance hobbed. Although Figure 86 shows that the averages for the two groups differed, statistical "t" tests indicated that these differences could have occurred by chance alone. (See Appendix III for description of "t" tests.) The comparisons were based on four data points in each set. Real differences would have to be very large to be detectable in such small samples. The results were therefore not inconsistent with the analysis of endurance limits which showed that, based on about 200 points, the fillet configuration does produce different endurance limits based on AGMA stresses. Even with this small sample, the results, while not conclusive, have the same sense as the more comprehensive analysis; i.e., protuberance hobbed fillet should produce a higher endurance limit when stresses are calculated with the AGMA formula.

TABLE XV
## RANKED ENDURANCE LIMITS FOR VARIOUS STRESS CALCULATION METHODS

| Configuration Number | Endurance Load (p. s. i.) | Lewis | | Heywood | | Kelley-Pedersa | |
|---|---|---|---|---|---|---|---|
| | | Configuration Number | Endurance Limit (p. s. i.) | Configuration Number | Endurance Limit (p. s. i.) | Configuration Number | End Limi |
| 10 | 96,429 | 16 | 154,560 | 5 | 164,050 | 9 | 162 |
| 4 | 94,968 | 6 | 143,040 | 9 | 162,182 | 13 | 149 |
| 11 | 90,107 | 15 | 138,530 | 13 | 150,419 | 5 | 145 |
| 16 | 88,149 | 13 | 123,070 | 15 | 148,948 | 11 | 143 |
| 9 | 86,978 | 10 | 122,610 | 11 | 148,539 | 15 | 142 |
| 15 | 83,507 | 4 | 122,250 | 7 | 137,582 | 1 | 133 |
| 12 | 80,647 | 7 | 118,660 | 1 | 134,517 | 7 | 113 |
| 13 | 74,698 | 5 | 116,430 | 6 | 107,429 | 6 | 91 |
| 6 | 72,192 | 14 | 116,360 | 10 | 94,267 | 10 | 89 |
| 14 | 65,807 | 9 | 115,035 | 4 | 87,820 | 4 | 88 |
| 7 | 65,698 | 11 | 115,000 | 16 | 82,852 | 16 | 75 |
| 2 | 64,400 | 8 | 110,210 | 3 | 74,769 | 12 | 64 |
| 1 | 61,901 | 12 | 100,080 | 2 | 74,000 | 3 | 74 |
| 8 | 60,622 | 1 | 90,562 | 12 | 70,617 | 2 | 74 |
| 5 | 59,165 | 2 | 88,754 | 14 | 69,581 | 14 | 65 |
| 3* | 42,689 | 3* | 58,292 | 8 | 67,914 | 8 | 52 |
| Average | 74,247 | | 114,590 | | 110,970 | | 104 |
| Range | 59,165 to 96,429 | | 90,562 to 154,560 | | 67,914 to 164,050 | | 52 162 |
| Variation = $\frac{\text{maximum}}{\text{minimum}}$ range = 1.63 | | | 1.71 | | 2.42 | | 3. |

Note: Configuration number 3 was deleted from range and variation calculation when it was lowest value.

| ...elley-Pedersen | | Dolan-Broghamer | | AGMA | | Test Rig Load | |
|---|---|---|---|---|---|---|---|
| Configuration Number | Endurance Limit (p. s. i.) | Configuration Number | Endurance Limit (p. s. i.) | Configuration Number | Endurance Limit (p. s. i.) | Configuration Number | Endurance Limit (p. s. i.) |
|  | 162,389 | 6 | 204,030 | 6 | 223,400 | 11 | 8,210 |
|  | 149,504 | 16 | 196,380 | 16 | 218,700 | 9 | 7,997 |
|  | 145,707 | 4 | 180,960 | 4 | 203,100 | 15 | 7,678 |
|  | 143,768 | 15 | 179,020 | 15 | 199,600 | 13 | 6,868 |
|  | 142,965 | 10 | 168,430 | 10 | 191,300 | 7 | 5,826 |
|  | 133,006 | 5 | 166,800 | 7 | 182,600 | 1 | 5,490 |
|  | 113,718 | 7 | 166,360 | 5 | 182,300 | 5 | 5,247 |
|  | 91,292 | 13 | 161,410 | 13 | 180,000 | 3 | 3,786 |
|  | 89,268 | 9 | 159,035 | 9 | 179,900 | 10 | 2,217 |
|  | 88,111 | 11 | 156,200 | 11 | 177,100 | 4 | 2,106 |
|  | 75,368 | 8 | 153,370 | 8 | 168,600 | 16 | 2,026 |
|  | 64,428 | 14 | 148,230 | 14 | 165,000 | 12 | 1,854 |
|  | 74,405 | 1 | 139,480 | 1 | 154,900 | 6 | 1,601 |
|  | 74,200 | 2 | 136,300 | 12 | 153,800 | 14 | 1,513 |
|  | 65,731 | 12 | 135,160 | 2 | 152,200 | 2 | 1,429 |
|  | 52,957 | 3* | 86,559 | 3* | 96,600 | 8 | 1,344 |
|  | 104,180 | | 158,600 | | 176,820 | | 4,075 |
|  | 52,957 to 162,389 | | 136,300 to 204,030 | | 153,800 to 223,400 | | 1,344 to 8,210 |
|  | 3.07 | | 1.50 | | 1.45 | | 6.11 |

B
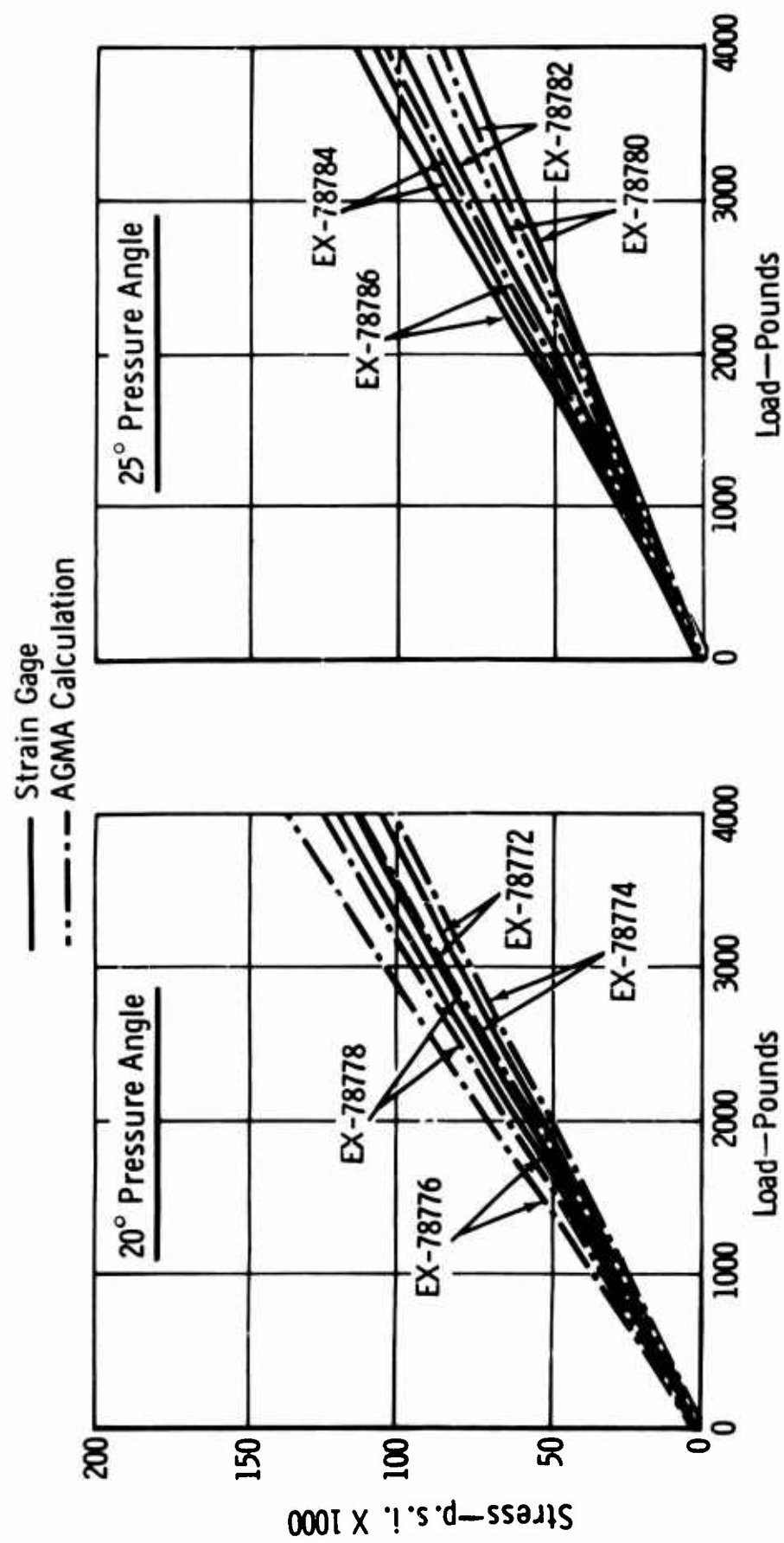
Figure 85. Calculated Stress for Gear Tooth Load.
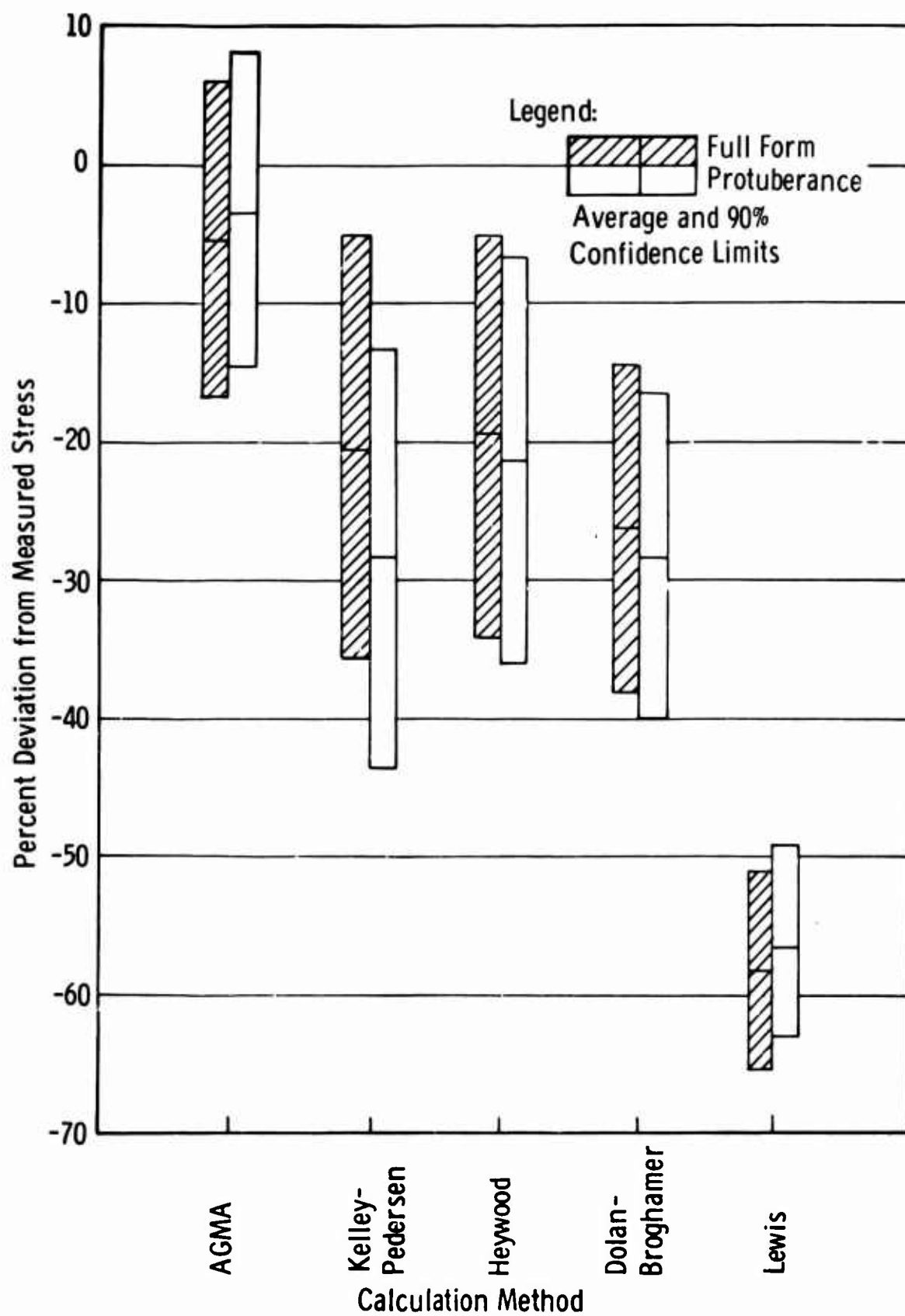
Figure 86. Comparison of Methods for Calculating Gear Stress.

TABLE XVI

GEAR CONFIGURATION RANKING COMPARISON

| Test Rig Load Ranking | Lewis | | Heywood | | Kelley-Pedersen | | Dolan-Broghamer | | AGMA | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Rank | Difference | Rank | Difference | Rank | Difference | Rank | Difference | Rank | Difference |
| 11 | 11 | 0 | 11 | 0 | 11 | 0 | 11 | 0 | 11 | 0 |
| 9 | 9 | 0 | 15 | 1 | 15 | 1 | 15 | 1 | 9 | 0 |
| 15 | 15 | 0 | 3 | 5 | 7 | 2 | 9 | 1 | 15 | 0 |
| 13 | 3 | 4 | 9 | 2 | 3 | 4 | 13 | 0 | 13 | 0 |
| 7 | 13 | 1 | 13 | 1 | 9 | 3 | 3 | 3 | 3 | 3 |
| 1 | 1 | 0 | 7 | 1 | 13 | 2 | 1 | 0 | 1 | 0 |
| 5 | 7 | 2 | 1 | 1 | 1 | 1 | 7 | 2 | 7 | 2 |
| 3 | 5 | 1 | 5 | 1 | 5 | 1 | 5 | 1 | 5 | 1 |
| 10 | 12 | 3 | 12 | 3 | 12 | 3 | 12 | 3 | 12 | 3 |
| 4 | 10 | 1 | 16 | 1 | 16 | 1 | 10 | 1 | 10 | 1 |
| 16 | 16 | 0 | 4 | 1 | 8 | 5 | 16 | 0 | 16 | 0 |
| 12 | 14 | 2 | 10 | 3 | 10 | 3 | 14 | 2 | 14 | 2 |
| 6 | 4 | 3 | 14 | 1 | 4 | 3 | 4 | 3 | 4 | 3 |
| 14 | 2 | 1 | 8 | 2 | 14 | 0 | 2 | 1 | 2 | 1 |
| 2 | 8 | 1 | 2 | 0 | 2 | 0 | 8 | 1 | 8 | 1 |
| 8 | 6 | 3 | 6 | 3 | 6 | 3 | 6 | 3 | 6 | 3 |
| Correct Rankings | 5 | | 2 | | 3 | | 4 | | 6 | |
| Prediction Accuracy | 1.375 | | 1.625 | | 2.000 | | 1.375 | | 1.25 | |

Note: Numbers in difference columns indicate difference in rank position from test rig load ranking.

TABLE XVII

FATIGUE TEST GEAR MEASURED DIMENSIONS

| Part Number | Serial Number | Dimension Over Pins | Root Diameter (inches) | Minimum Root Radius (inch) | Tooth Edge Break (inch)‡ | | | |
|---|---|---|---|---|---|---|---|---|
| | | | | | 1 | 2 | 3 | 4 |
| EX-78772 | CX 9089 | 4.3926 | 3.5806 | 0.065 | — | — | — | — |
| | CX 9090 | 4.3938 | 3.5800 | 0.065 | 0.020 CUS | 0.020 CUD | 0.020 CUD | 0.020 CUD |
| | CX 9091 | 4.3947 | 3.5800 | 0.065 | 0.020 CE | 0.015 CE | 0.015 CE | 0.015 CE |
| | CX 9092 | 4.3950 | 3.5802 | 0.065 | 0.010 CUS | 0.020 CUS | 0.010 CUS | 0.020 CUD |
| | CX 9093 | 4.3950 | 3.5830 | 0.065 | — | — | — | — |
| EX-78773 | CX 9074 | 2.1920 | 1.7835 | 0.040 | 0.005 CE | 0.005 CE | 0.005 CE | 0.005 CE |
| | CX 9075 | 2.1921 | 1.7838 | 0.040 | 0.005 CE | 0.005 CE | 0.005 CE | 0.005 CE |
| | CX 9076 | 2.1942 | 1.7903 | 0.038 | 0.005 RE | 0.005 RE | 0.005 RE | 0.005 RE |
| | CX 9077 | 2.1920 | 1.7838 | 0.040 | — | — | — | — |
| | CX 9078 | 2.1922 | 1.7836 | 0.040 | 0.005 CE | — | — | — |
| EX-78774 | CX 9064 | 4.3981 | 3.5875 | 0.070 | 0.030 CE | 0.020 CE | 0.020 CE | 0.020 CE |
| | CX 9065 | 4.3976 | 3.5867 | 0.070 | 0.020 CUS | 0.030 CE | 0.020 CUS | 0.030 CUS |
| | CX 9066 | 4.3981 | 3.5863 | 0.070 | — | — | — | — |
| | CX 9067 | 4.3947 | 3.5820 | 0.070 | — | 0.030 CE | 0.030 CE | 0.030 CE |
| | CX 9068 | 4.3984 | 3.5882 | 0.070 | 0.030 CE | 0.020 CUS | 0.020 CUD | 0.020 CUS |
| EX-78775 | CX 9099 | 2.1912 | 1.7955 | 0.034 | S | S | S | S |
| | CX 9100 | 2.1930 | 1.7955 | 0.034 | — | — | — | — |
| | CX 9101 | 2.1918 | 1.7955 | 0.035 | — | — | — | — |
| | CX 9102 | 2.1926 | 1.7955 | 0.035 | S | 0.005 RE | S | S |
| | CX 9103 | 2.1897 | 1.7955 | 0.024 | — | — | — | — |
| EX-78776 | CX 9007 | 4.3965 | 3.5252 | 0.068 | 0.030 CUS | — | — | — |
| | CX 9008 | 4.3964 | 3.5231 | 0.068 | 0.030 CUS | 0.030 RE | 0.030 CUS | 0.030 CUS |
| | CX 9009 | 4.3963 | 3.5237 | 0.068 | 0.015 CE | — | — | 0.015 CUD |
| | CX 9010 | 4.3963 | 3.5241 | 0.068 | S | 0.010 CUS | 0.010 CUS | 0.015 CUS |
| | CX 9011 | 4.3964 | 3.5227 | 0.068 | — | — | — | — |

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**TABLE XVII (CONT)**

| Part Number | Serial Number | Dimension Over Pins | Root Diameter (inches) | Minimum Root Radius (inch) | Tooth Edge Break (inch)* | | | |
|---|---|---|---|---|---|---|---|---|
| | | | | | 1 | 2 | 3 | 4 |
| EX-78777 | CX 9059 | 2.1967 | 1.767 | 0.030 | — | — | — | — |
| | CX 9060 | 2.1967 | 1.768 | 0.030 | — | — | — | — |
| | CX 9061 | 2.1967 | 1.767 | 0.032 | 0.020 CE | 0.020 CE | 0.020 CE | 0.020 CE |
| | CX 9062 | 2.1967 | 1.769 | 0.032 | 0.020 CE | 0.020 CE | 0.020 CE | 0.020 CE |
| | CX 9063 | 2.1967 | 1.767 | 0.032 | 0.020 CE | 0.020 CE | 0.020 CE | 0.020 CE |
| EX-78778 | CX 9054 | 4.3904 | 3.5267 | 0.090 | 0.030 CUD | 0.030 CUD | 0.030 CUD | 0.030 CE |
| | CX 9055 | 4.3905 | 3.5267 | 0.090 | — | — | — | — |
| | CX 9056 | 4.3903 | 3.5266 | 0.090 | 0.010 CE | 0.020 CE | 0.020 CE | 0.020 CUS |
| | CX 9057 | 4.3905 | 3.5275 | 0.090 | 0.030 CE | 0.030 CE | 0.030 CUD | 0.030 CUS |
| | CX 9058 | 4.3906 | 3.5281 | 0.090 | — | — | — | — |
| EX-78779 | CX 9104 | 2.1961 | 1.7678 | 0.044 | — | — | — | — |
| | CX 9105 | 2.1961 | 1.7672 | 0.044 | 0.020 CE | 0.020 CE | 0.020 CE | 0.020 CUD |
| | CX 9106 | 2.1961 | 1.7679 | 0.044 | 0.020 CE | 0.020 CE | 0.020 CE | 0.020 CE |
| | CX 9107 | 2.1963 | 1.7682 | 0.044 | — | — | — | — |
| | CX 9108 | 2.1963 | 1.7680 | 0.044 | 0.020 CE | — | — | — |
| EX-78780 | CX 9094 | 4.3979 | 3.6000 | 0.055 | 0.020 CE | 0.020 CE | 0.020 CE | 0.020 CE |
| | CX 9095 | 4.3978 | 3.5999 | 0.055 | 0.030 CE | 0.030 CE | 0.030 CE | 0.030 CE |
| | CX 9096 | 4.3978 | 3.5995 | 0.055 | 0.020 RE | 0.020 RE | 0.020 RE | 0.020 RE |
| | CX 9097 | 4.3980 | 3.6005 | 0.055 | — | — | — | — |
| | CX 9098 | 4.3985 | 3.5998 | 0.055 | — | — | — | — |
| EX-78781 | CX 9030 | 2.1968 | 1.8097 | 0.028 | 0.010 CUD | 0.010 CE | 0.005 CE | 0.020 CE |
| | CX 9031 | 2.1967 | 1.8095 | 0.028 | 0.010 CE | 0.020 CE | 0.010 CE | 0.010 CE |
| | CX 9032 | 2.1976 | 1.8105 | 0.026 | 0.005 CE | 0.005 CE | 0.005 CE | 0.005 CE |
| | CX 9033 | 2.1961 | 1.8093 | 0.026 | 0.005 CE | 0.005 CE | 0.005 CE | 0.005 CE |
| | CX 9034 | 2.1969 | 1.8096 | 0.028 | 0.010 CE | 0.005 CE | 0.005 CE | 0.005 CUD |
| EX-78782 | CX 9109 | 4.3967 | 3.6050 | 0.070 | — | — | — | — |
| | CX 9110 | 4.3976 | 3.6035 | 0.070 | — | — | — | — |
| | CX 9111 | 4.3976 | 3.6040 | 0.070 | 0.010 CUD | 0.020 CE | 0.020 CE | 0.020 CUD |
| | CX 9112 | 4.3978 | 3.6040 | 0.070 | 0.015 CE | 0.010 CE | 0.010 CE | 0.005 CE |
| | CX 9113 | 4.3981 | 3.6035 | 0.070 | — | — | — | — |

**TABLE XVII** (CONT)

| Part Number | Serial Number | Dimension Over Pins | Root Diameter (inches) | Minimum Root Radius (inch) | Tooth Edge Break (inch)* 1 | 2 | 3 | 4 |
|---|---|---|---|---|---|---|---|---|
| EX-78783 | CX 9025 | 2.1965 | 1.805 | 0.036 | 0.020 CE | 0.020 CE | 0.020 CE | 0.020 CE |
|  | CX 9026 | 2.1972 | 1.805 | 0.036 | 0.020 CE | 0.020 CE | 0.020 CE | 0.020 CE |
|  | CX 9027 | 2.1967 | 1.805 | 0.036 | 0.020 CE | 0.020 CE | 0.020 CE | 0.020 CE |
|  | CX 9028 | 2.1947 | 1.803 | 0.034 | 0.020 CE | 0.020 CE | 0.020 CE | 0.020 CE |
|  | CX 9029 | 2.1968 | 1.805 | 0.036 | 0.020 CE | 0.020 CUD | 0.020 CUD | 0.020 CE |
| EX-78784 | CX 9069 | 4.3980 | 3.5424 | 0.065 | — | — | — | — |
|  | CX 9070 | 4.3974 | 3.5415 | 0.065 | 0.030 CUD | 0.030 CUD | 0.030 CUS | 0.020 CUS |
|  | CX 9071 | 4.3975 | 3.5413 | 0.065 | 0.030 CE | 0.020 CUD | 0.030 CUD | 0.020 CUD |
|  | CX 9072 | 4.3975 | 3.5418 | 0.065 | 0.020 CUD | 0.020 CUD | 0.020 CUD | 0.030 CUD |
|  | CX 9073 | 4.3973 | 3.5412 | 0.065 | 0.005 CE | 0.005 CE | 0.005 CE | 0.005 CE |
| EX-78785 | CX 9035 | 2.1975 | 1.775 | 0.033 | 0.020 CE | 0.020 CE | 0.020 CE | 0.020 CE |
|  | CX 9036 | 2.1976 | 1.776 | 0.033 | — | — | — | — |
|  | CX 9037 | 2.1972 | 1.775 | 0.033 | 0.020 CE | 0.020 CE | 0.020 CE | 0.020 CE |
|  | CX 9038 | 2.1978 | 1.776 | 0.033 | 0.010 CUD | 0.005 CUS | 0.010 CUS | 0.010 CUD |
|  | CX 9039 | 2.1974 | 1.776 | 0.033 | — | — | — | — |
| EX-78786 | CX 9012 | 4.3982 | 3.5436 | 0.073 | — | — | — | — |
|  | CX 9013 | 4.3983 | 3.5445 | 0.073 | 0.050 CE | 0.050 CE | 0.040 CE | 0.040 CE |
|  | CX 9014 | 4.3983 | 3.5440 | 0.073 | 0.030 CE | 0.020 CUD | 0.020 CE | 0.030 CE |
|  | CX 9015 | 4.3983 | 3.5448 | 0.073 | 0.030 CE | 0.030 CE | 0.030 CE | 0.030 CUD |
|  | CX 9016 | 4.3982 | 3.5447 | 0.073 | — | — | — | — |
| EX-78787 | CX 9114 | 2.1947 | 1.7785 | 0.036 | 0.020 CE | 0.020 CE | 0.020 CE | 0.020 CE |
|  | CX 9115 | 2.1945 | 1.7785 | 0.034 | 0.020 CE | 0.020 CE | 0.020 CE | 0.020 CE |
|  | CX 9116 | 2.1945 | 1.7785 | 0.034 | 0.020 CE | 0.020 CE | 0.020 CE | 0.020 CE |
|  | CX 9117 | 2.1949 | 1.7785 | 0.034 | 0.020 CE | 0.020 CE | 0.020 CE | 0.020 CE |
|  | CX 9118 | 2.1946 | 1.7785 | 0.036 | 0.020 CE | 0.020 CE | 0.020 CE | 0.020 CE |

*Note—Edge Break Code:
C—edge break approximates chamfer
R—edge break approximates radius
E—even blend from flank to root
UD—uneven but rounded blend from flank to root
US—uneven blend leaving sharp edge at weakest section
S—sharp

TABLE XVIII
MEASURED STRESS OF FATIGUE TEST GEARS COMPARED
WITH CALCULATED STRESS

| Fatigue Test Gear | Pitch | Pressure Angle (degrees) | Fillet Radius (inch) | Fillet Configuration | Measured Strain Gage Strain Rate* | AGMA Strain Rate* | Percent Deviation | Kelley-Pedersen Strain Rate* | Perc Devia† |
|---|---|---|---|---|---|---|---|---|---|
| EX-78772 | 6 | 20 | 0.050 | Full form | 927 | 941 | + 1.5 | 810 | -12. |
| EX-78774 | 6 | 20 | 0.080 | Full form | 1010 | 850 | -15.8 | 655 | -35. |
| EX-78776 | 6 | 20 | 0.050 | Protuberance | 1150 | 1157 | + 0.6 | 923 | -19. |
| EX-78778 | 6 | 20 | 0.080 | Protuberance | 1008 | 1042 | + 3.4 | 652 | -35. |
| EX-78780 | 6 | 25 | 0.050 | Full form | 691 | 750 | + 8.5 | 677 | - 2. |
| EX-78782 | 6 | 25 | 0.067 | Full form | 856 | 718 | -16.1 | 584 | -31. |
| EX-78784 | 6 | 25 | 0.050 | Protuberance | 900 | 873 | - 3.0 | 723 | -19. |
| EX-78786 | 6 | 25 | 0.067 | Protuberance | 1017 | 867 | -14.5 | 621 | -39. |

*Strain Rate— · hes/inch/1000 pounds

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A

| Petersen Rate[*] | Percent Deviation | Heywood Strain Rate[*] | Percent Deviation | Dolan-Broghamer Strain Rate[*] | Percent Deviation | Lewis Strain Rate[*] | Percent Deviation | Stress Concentration Factor (Lewis) |
|---|---|---|---|---|---|---|---|---|
| | -12.6 | 817 | -11.9 | 756 | -18.5 | 423 | -54.5 | 2.19 |
| | -35.1 | 659 | -34.8 | 645 | -36.1 | 367 | -63.6 | 2.75 |
| | -19.7 | 1040 | - 9.6 | 945 | -17.8 | 591 | -48.6 | 1.95 |
| | -35.4 | 787 | -21.9 | 760 | -22.6 | 466 | -53.8 | 2.16 |
| | - 2.2 | 675 | - 2.3 | 585 | -15.4 | 328 | -52.5 | 2.10 |
| | -31.8 | 602 | -29.7 | 555 | -35.2 | 314 | -63.3 | 2.72 |
| | -19.7 | 730 | -18.9 | 622 | -30.8 | 367 | -59.2 | 2.45 |
| | -39.0 | 646 | -36.5 | 585 | -42.5 | 349 | -65.6 | 2.91 |

B

In summary, the bar chart in Figure 87 shows the average degree of correlation for the various methods of calculation versus the measured data. It is apparent that the AGMA method offers the greatest degree of correlation.

PHOTOSTRESS DATA

As described in the section titled Results, the photostress investigations showed the stress location and stress distribution to be in agreement with the theoretical location.

EFFECT OF GEOMETRIC VARIABLES OF GEAR FATIGUE TEST

The following studies of the data evaluate the four variables of the gear fatigue test. Despite the high precision achieved in the manufacture of test gears, the scatter in fatigue life was high. Many run-outs (termination of test before failure) occurred, although the planned stress levels were altered in an attempt to fail teeth with $10^7$ cycles. It was decided, therefore, to base the analysis on the endurance limit produced by each of the 16 configurations of gear teeth by developing a mathematical model for the S/N curve. The derivation of the analytical model is included in Appendix V. This method was used to determine the characteristic and fit of the S/N curve for all the fatigue test points, stress curves, and R. R. Moore curves. S/N curves were fitted to the gear tooth fatigue data with respect to basic applied load, AGMA calculated stress, and Kelley-Pedersen calculated stress. The basic applied load (test rig load) was used as a positive baseline since it is unaffected by any calculations. The AGMA calculated stress was of prime interest, since it was determined to be the best predictive calculation method. The Kelley-Pedersen method was used as a second stress method to provide direct comparison for the AGMA method. The endurance limits obtained from the S/N curves were used to evaluate each of the four geometric variables and their interactions—i.e., diametral pitch, pressure angle, fillet size, and fillet configuration.
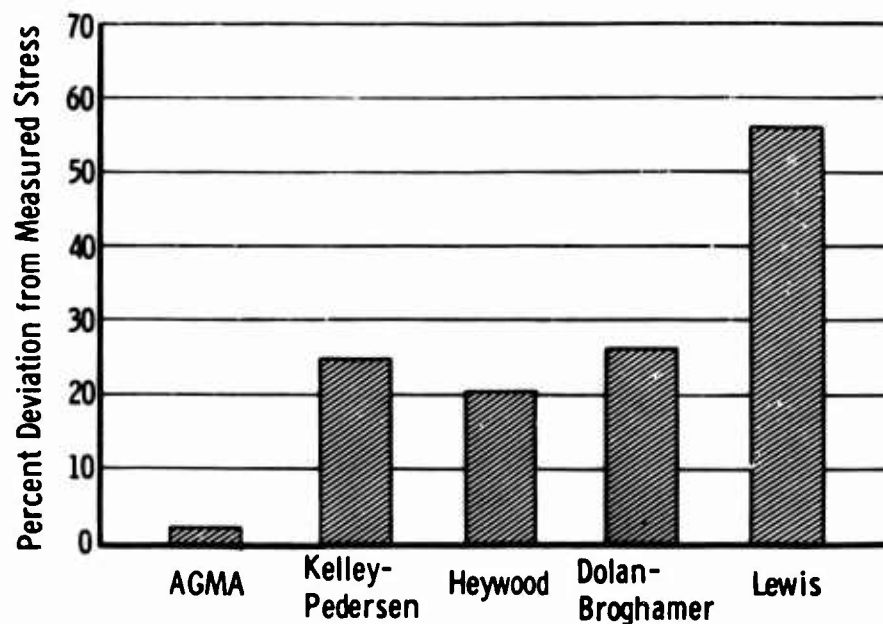


Figure 87. Comparison of Calculated and Measured Stresses.

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A summary of significant test results is given in the following paragraphs. The pre-selected significance level was $a = 0.05$, which corresponds to a statistical "t" value of 2.0. This level indicates that the result would occur 95 out of 100 times. A discussion of the statistical test of significance is included in Appendix III.

## Diametral Pitch

As would be expected, due to the different face width and pitch, a significant effect was found for diametral pitch (6 and 12) based on applied load. It would be expected that stress calculations would adequately consider these geometric variables. It was found that the AGMA stress calculation did adequately predict a stress level. The Kelley-Pedersen method reduced the significance value but was still very significant. Table XIX summarizes these data (the load values have been corrected for diametral pitch and load for comparison).

TABLE XIX
EFFECT OF DIAMETRAL PITCH ON GEAR FATIGUE DATA

| Diametral Pitch | Load (pounds) | Corrected Load (pounds)* | AGMA Stress (p.s.i.) | Kelley-Pedersen Stress (p.s.i.) |
|---|---|---|---|---|
| 6 | 6674 | 6807** | 175,500 | 138,750 |
| 12 | 1795 | 6820 | 184,600 | 75,500 |

*Corrected 12 pitch as follows for comparison with 6 pitch on a load basis:

| | Pitch | 6 | 12 | Correction |
|---|---|---|---|---|
| Pitch | | 6 | 12 | 2.00 x load |
| Face Width (inch) | | 0.500 | 0.250 | 2.00 x load |
| | | | | Total = 4.0 x Load |
| Y (average) | | 0.513 | 0.486 | 0.95 x 4.0 x load = 3.8 x load |

3.8 x 1795 = 6820 pounds

**Per reference 37, a 2-percent size effect might be expected for the range of face widths tested; therefore, 1.02 x 6674 = 6807 pounds.

## Pressure Angle

A significant effect was found due to the change in 20- and 25-degree pressure angle gears based on applied load. Also, it would be expected that the stress calculation should adequately predict this geometric effect. The study indicated that the AGMA and Kelley-Pedersen calculation methods adequately predicted the stress level. Table XX summarizes these data (the load values have been corrected for pressure angle for comparison).

TABLE XX
EFFECT OF PRESSURE ANGLE ON GEAR FATIGUE DATA

| Pressure Angle (degrees) | Load (pounds) | Corrected Load (pounds)* | AGMA Stress (p.s.i.) | Kelley-Pedersen Stress (p.s.i.) |
|---|---|---|---|---|
| 20 | 3802 | 5027 | 176,500 | 104,480 |
| 25 | 4328 | 4328 | 183,600 | 105,700 |

*Correction for pressure angle was made by averaging Y values for 20- and 25-degree pressure angle gears.

20-degree average Y = 0.4302
25-degree average Y = 0.5688 $\qquad 3802 \times \dfrac{0.5688}{0.4302} = 5027$ pounds

## Fillet Size

For the practical range of fillet sizes tested, no significant difference was found on the basis of applied load or AGMA calculations. A significant difference was found, however, on the basis of the Kelley-Pedersen calculated stress. These data are summarized as follows:

|              | Load (pounds) | AGMA Stress (p. s. i.) | Kelley-Pedersen Stress (p. s. i.) |
|--------------|---------------|------------------------|-----------------------------------|
| Small Fillet | 3915          | 179,000                | 111,960                           |
| Large Fillet | 4246          | 181,500                | 98,540                            |

## Fillet Configuration

For the fillet configurations tested—full form and protuberance hobbed—no significant difference was found on the basis of applied load or the Kelley-Pedersen method. A significant difference was found, however, on the basis of calculated AGMA stress. These data are summarized as follows:

|              | Load (pounds) | AGMA Stress (p. s i.) | Kelley-Pedersen Stress (p. s. i.) |
|--------------|---------------|-----------------------|-----------------------------------|
| Full form    | 4234          | 169,300               | 106,100                           |
| Protuberance | 3908          | 193,000               | 104,100                           |

The average endurance limit for each variable and the corresponding statistical "t" value for the tests of significance are presented in Table XXI. Several interactions were found, as indicated in the table.

It is apparent that the AGMA formula adequately predicts gear tooth bending stress with but two exceptions: fillet configuration and the interaction of pressure angle, fillet radius, and fillet configuration. No exact reason for these differences can be shown. The difference may be due to any of the changes previously listed between the two fillet configurations such as residual stress, case depth, surface finish, etc. In view of the interaction obtained and its relative value, the difference may be due to the accumulation of errors in extrapolation of the stress concentration factor.

The significant differences between levels for each factor are apparent. Changing the value assigned to any significant geometric factor produces a change in the endurance limit. This limit is larger than can be explained by the inherent variability associated with fatigue testing. For example, diametral pitch was significant in terms of basic load, as was expected. The reduction in endurance limit in going from a diametral pitch of 6 to 12 was 4879 pounds. The fillet configuration was not significant in terms of basic load; the difference between endurance limits for the full form and the protuberance configuration was only 326 pounds.

The interpretation of significant interactions is more difficult. In general, it can be stated that the change in endurance limits caused by changing one factor is dependent on the value assigned to the interacting factor. An example is provided by the significant AB interaction associated with applied load. See Table XXI. At the 20-degree pressure angle, the endurance limit is reduced from 5780 to 1610 pounds in going from a diametral pitch value of 6 to 12; at the 25-degree pressure angle, the endurance limit

## TABLE XXI
## ANALYSIS OF GEOMETRIC VARIABLES AND INTERACTIONS

| | Basic Applied Load (pounds) | | | AGMA Stress (p.s.i. × 1000) | | | Kelley-Pedersen Stress (p.s.i. × 1000) | | |
|---|---|---|---|---|---|---|---|---|---|
| | Low | High | t* | Low | High | t* | Low | High | t* |
| A—Diametral Pitch | 6674 | 1795 | **24.27 | 175,500 | 184,600 | 1.15 | 138,750 | 75,500 | **13.8 |
| B—Pressure Angle | 3802 | 4328 | ** 2.62 | 176,500 | 183,600 | 0.89 | 104,480 | 105,700 | 0.3 |
| C—Fillet Size | 3915 | 4246 | 1.65 | 179,000 | 181,500 | 0.31 | 111,960 | 98,540 | ** 2.9 |
| D—Fillet Configuration | 4234 | 3908 | 1.62 | 168,300 | 193,000 | **2.98 | 106,100 | 104,100 | 0.4 |
| AB Interaction | 3620 | 4660 | ** 5.17 | 176,100 | 185,500 | 1.18 | 97,800 | 114,300 | ** 3.6 |
| AC Interaction | 3975 | 4187 | 1.05 | 180,500 | 180,200 | 0.03 | 106,040 | 104,300 | 0.4 |
| AD Interaction | 4286 | 3878 | ** 2.03 | 182,900 | 178,200 | 0.53 | 105,910 | 104,390 | 0.3 |
| BC Interaction | 3765 | 4430 | ** 3.31 | 182,800 | 177,600 | 0.65 | 102,360 | 108,200 | 1.3 |
| BD Interaction | 4195 | 4003 | 0.96 | 179,100 | 181,700 | 0.20 | 105,900 | 104,320 | 0.3 |
| CD Interaction | 4024 | 4139 | 0.57 | 183,400 | 177,400 | 0.76 | 103,750 | 106,490 | 0.6 |
| ABC Interaction | 4006 | 4181 | 0.87 | 180,000 | 180,600 | 0.08 | 103,000 | 107,910 | 1.7 |
| ABD Interaction | 3999 | 4154 | 0.77 | 183,900 | 177,200 | 0.84 | 104,950 | 105,330 | 0.1 |
| ACD Interaction | 4106 | 4057 | 0.24 | 179,900 | 180,800 | 0.11 | 105,560 | 104,700 | 0.2 |
| BCD Ineraction | 4307 | 3828 | ** 2.38 | 193,900 | 165,000 | **3.64 | 112,160 | 97,220 | ** 3.3 |

*t = statistical "t" test of significant value.

**Denotes significance at α = 0.05.

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is reduced from 7650 to 1930 pounds for the same change in diametral pitch. This example is shown graphically in Figure 88. The interaction is indicated by the convergence of the lines; i. e., the difference in endurance limits between a 20- and a 25-degree pressure angle is not the same at the two values of diametral pitch. The information used is presented in Tables XXII, XXIII, and XXIV for the basic applied load and the AGMA and Kelley-Pedersen calculated stress.

(1) LOAD—Diametral Pitch and Pressure Angle
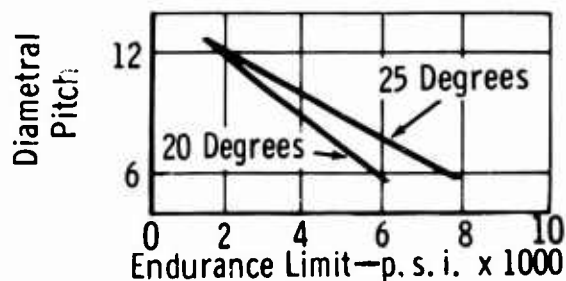


(2) LOAD—Diametral Pitch and Fillet Configuration
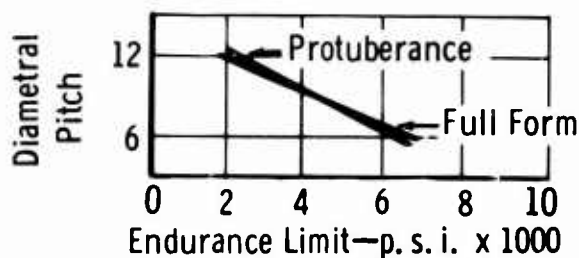


(3) LOAD—Pressure Angle and Root Fillet Radius
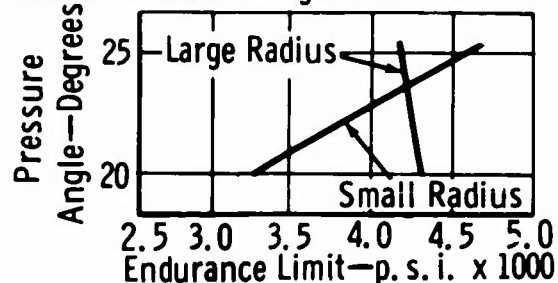


(4) KELLEY-PEDERSEN—Diametral Pitch and Pressure Angle
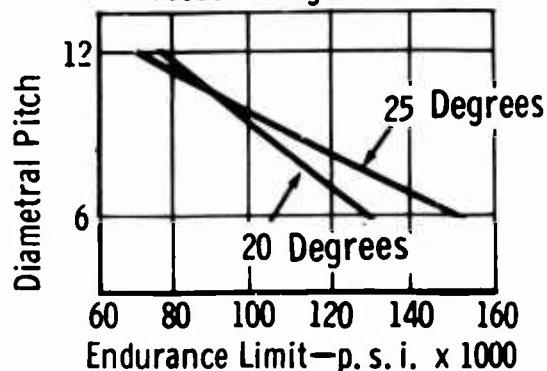


Figure 88. Significant Two-Factor Interactions.

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## TABLE XXII

## ENDURANCE LIMITS BASED ON BASIC GEAR TOOTH LOADING

| Root Radius | Fillet Configuration | | Diametral Pitch | | | |
|---|---|---|---|---|---|---|
| | | | 6 | | 12 | |
| | | | Pressure Angle (degrees) | | | |
| | | | 20 | 25 | 20 | 25 |
| Small | Full Form | 1* | 1 = EX-78772 | 9 = EX-78780 | 2 = EX-78773 | 10 = EX-78781 |
| | | 2** | 12 | 12 | 14 | 16 |
| | | 3† | $5.490 \times 10^3$ | $7.995 \times 10^3$ | $1.429 \times 10^3$ | $2.217 \times 10^3$ |
| | | 4‡ | $3.080 \times 10^5$ | $3.476 \times 10^5$ | $0.3543 \times 10^5$ | $0.0749 \times 10^5$ |
| | Protuberance | 1 | 5 = EX-78776 | 13 = EX-78784 | 6 = EX-78777 | 14 = EX-78785 |
| | | 2 | 11 | 13 | 13 | 12 |
| | | 3 | $5.247 \times 10^3$ | $6.867 \times 10^3$ | $1.601 \times 10^3$ | $1.513 \times 10^3$ |
| | | 4 | $1.805 \times 10^5$ | $1.170 \times 10^5$ | $0.0207 \times 10^5$ | $0.5195 \times 10^5$ |
| Large | Full Form | 1 | 3 = EX-78774 | 11 = EX-78782 | 4 = EX-78775 | 12 = EX-78783 |
| | | 2 | 19 | 10 | 10 | 19 |
| | | 3 | $6.238 \times 10^3$ | $8.210 \times 10^3$ | $2.105 \times 10^3$ | $1.854 \times 10^3$ |
| | | 4 | $0.4042 \times 10^5$ | $5.832 \times 10^5$ | $0.1131 \times 10^5$ | $1.134 \times 10^5$ |
| | Protuberance | 1 | 7 = EX-78778 | 15 = EX-78786 | 8 = EX-78779 | 16 = EX-78787 |
| | | 2 | 9 | 12 | 9 | 18 |
| | | 3 | $5.827 \times 10^3$ | $7.678 \times 10^3$ | $1.344 \times 10^3$ | $2.026 \times 10^3$ |
| | | 4 | $4.934 \times 10^5$ | $0.0727 \times 10^5$ | $0.3142 \times 10^5$ | $0.0461 \times 10^5$ |

*1—Configuration number and part number.
**2—Sample size (data points) used to compute endurance limit.
†3—Endurance limit, pounds.
‡4—Variance of endurance limit, pounds.

TABLE XXIII

ENDURANCE LIMITS BASED ON AGMA CALCULATED STRESS

| Root Radius | Fillet Configuration | | Diametral Pitch | | | |
|---|---|---|---|---|---|---|
| | | | 6 | | 12 | |
| | | | Pressure Angle (degrees) | | | |
| | | | 20 | 25 | 20 | 25 |
| Small | Full Form | 1* | 1 = EX-78772 | 9 = EX-78780 | 2 = EX-78773 | 10 = EX-78781 |
| | | 2** | 12 | 12 | 14 | 16 |
| | | 3† | $1.549 \times 10^5$ | $1.798 \times 10^5$ | $1.524 \times 10^5$ | $1.913 \times 10^5$ |
| | | 4‡ | $2.3966 \times 10^8$ | $1.7827 \times 10^8$ | $4.252 \times 10^8$ | $0.535 \times 10^8$ |
| | Protuberance | 1 | 5 = EX-78776 | 13 = EX-78784 | 6 = EX-78777 | 14 = EX-78785 |
| | | 2 | 11 | 13 | 13 | 12 |
| | | 3 | $1.823 \times 10^5$ | $1.800 \times 10^5$ | $2.234 \times 10^5$ | $1.650 \times 10^5$ |
| | | 4 | $2.1788 \times 10^8$ | $0.8038 \times 10^8$ | $0.3870 \times 10^8$ | $6.1814 \times 10^8$ |
| Large | Full Form | 1 | 3 = EX-78774 | 11 = EX-78782 | 4 = EX-78775 | 12 = EX-78783 |
| | | 2 | 19 | 10 | 10 | 19 |
| | | 3 | $1.592 \times 10^5$ | $1.771 \times 10^5$ | $2.031 \times 10^5$ | $1.538 \times 10^5$ |
| | | 4 | $0.8754 \times 10^8$ | $2.7160 \times 10^8$ | $0.9998 \times 10^8$ | $7.8283 \times 10^8$ |
| | Protuberance | 1 | 7 = EX-78778 | 15 = EX-78786 | 8 = EX-78779 | 16 = EX-78787 |
| | | 2 | 9 | 12 | 9 | 18 |
| | | 3 | $1.826 \times 10^5$ | $1.996 \times 10^5$ | $1.686 \times 10^5$ | $2.187 \times 10^5$ |
| | | 4 | $4.6733 \times 10^8$ | $0.0491 \times 10^8$ | $4.9309 \times 10^8$ | $0.5281 \times 10^8$ |

*1—Configuration number and part number.
**2—Sample size (data points) used to compute endurance limit.
†3—Endurance limit, p.s.i.
‡4—Variance of endurance limit, p.s.i.

## TABLE XXIV

### ENDURANCE LIMITS BASED ON KELLEY-PEDERSEN CALCULATED STRESS

| Root Radius | Fillet Configuration | | Diametral Pitch | | | |
|---|---|---|---|---|---|---|
| | | | 6 | | 12 | |
| | | | Pressure Angle (degrees) | | | |
| | | | 20 | 25 | 20 | 25 |
| Small | Full Form | 1* | 1 = EX-78772 | 9 = EX-78780 | 2 = EX-78773 | 10 = EX-78781 |
| | | 2** | 12 | 12 | 14 | 16 |
| | | 3† | $133.03 \times 10^3$ | $162.34 \times 10^3$ | $74.11 \times 10^3$ | $89.29 \times 10^3$ |
| | | 4‡ | $17.67 \times 10^7$ | $14.52 \times 10^8$ | $8.31 \times 10^7$ | $1.17 \times 10^7$ |
| | Protuberance | 1 | 5 = EX-78776 | 13 = EX-78784 | 6 = EX-78777 | 14 = EX-78785 |
| | | 2 | 11 | 13 | 13 | 12 |
| | | 3 | $145.68 \times 10^3$ | $149.47 \times 10^3$ | $91.31 \times 10^3$ | $65.73 \times 10^3$ |
| | | 4 | $13.92 \times 10^7$ | $5.54 \times 10^7$ | $0.65 \times 10^7$ | $9.81 \times 10^7$ |
| Large | Full Form | 1 | 3 = EX-78774 | 11 = EX-78782 | 4 = EX-78775 | 12 = EX-78783 |
| | | 2 | 19 | 10 | 10 | 19 |
| | | 3 | $122.62 \times 10^3$ | $143.76 \times 10^3$ | $88.10 \times 10^3$ | $64.44 \times 10^3$ |
| | | 4 | $4.712 \times 10^7$ | $17.90 \times 10^7$ | $1.88 \times 10^7$ | $13.74 \times 10^7$ |
| | Protuberance | 1 | 7 = EX-78778 | 15 = EX-78786 | 8 = EX-78779 | 16 = EX-78787 |
| | | 2 | 9 | 12 | 9 | 18 |
| | | 3 | $113.72 \times 10^3$ | $143.01 \times 10^3$ | $52.97 \times 10^3$ | $75.35 \times 10^3$ |
| | | 4 | $18.12 \times 10^7$ | $0.25 \times 10^7$ | $4.86 \times 10^7$ | $0.63 \times 10^7$ |

*1—Configuration number and part number.
**2—Sample size (data points) used to compute endurance limit.
†3—Endurance limit, p.s.i.
‡4—Variance of endurance limit, p.s.i.

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The endurance limit for test gear configuration number 1 (EX-78774) was increased from a computed 96,600-p.s.i. AGMA stress value to 159,200 p.s.i. It was necessary to neutralize this low value to prevent bias to the designed experiment. The new value was determined by proportioning the configuration number 1 endurance limit based on fillet size. Fillet size is the only difference between configurations 1 and 3. The basic applied load and Kelley-Pedersen endurance limit for configuration 3 were similarly proportioned.

## BASIC MATERIAL STRENGTH

An ideal bending stress calculation would permit direct correlation of tooth strength with the basic material strength. R. R. Moore rotating beam fatigue test data were compared with fatigue test gear data to determine the degree of correlation.

The R. R. Moore S/N curve shown in Figure 89 presents the basic bending strength of the carburized AMS-6265 material of the test gears. R. R. Moore rotating beam specimens are related to gears as described in the following paragraphs.

### Type of Loading

The R. R. Moore test bar rotates while supporting a bending load. This results in complete reversal of the bending load on the test bar once each revolution. The relationship of fatigue data for the two types of loading is indicated in the modified Goodman diagram in Figure 90. Metallurgical investigations showed that the fatigue failures for the R. R. Moore samples and the test gears started on the carburized case surface. The modified Goodman diagram, therefore, is based on the case material properties. The ultimate strength level for the case was calculated by increasing the measured ultimate strength of the core material by the ratio of the case hardness and the core hardness at the surface:

$$180,000 \times \frac{58}{38} = 274,000 \text{ p.s.i.}$$

Points A and B in Figure 89 are located on the S/N curve to establish $10^8$ and $10^5$ cycle lines. These points are then plotted on the modified Goodman diagram, Figure 90, at the zero mean stress ordinate. Since the gear tooth load was in one direction only, the one-direction line was drawn at a slope of 2. A slope of 2 is used since the mean stress is one-half of the maximum stress for one-direction loading as shown in the following sketch. The intersection of the one-direction line and the cycle lines,



R. R. Moore
Completely Reversed

Gear Fatigue Test
One Direction

points C and D, establish points for an R. R. Moore S/N data curve modified for the fatigue test gear mode of loading. The modified S/N curve is shown in Figure 89. This modification is not required for use with idler gear applications where the gear tooth is subjected to complete reversal of loading.

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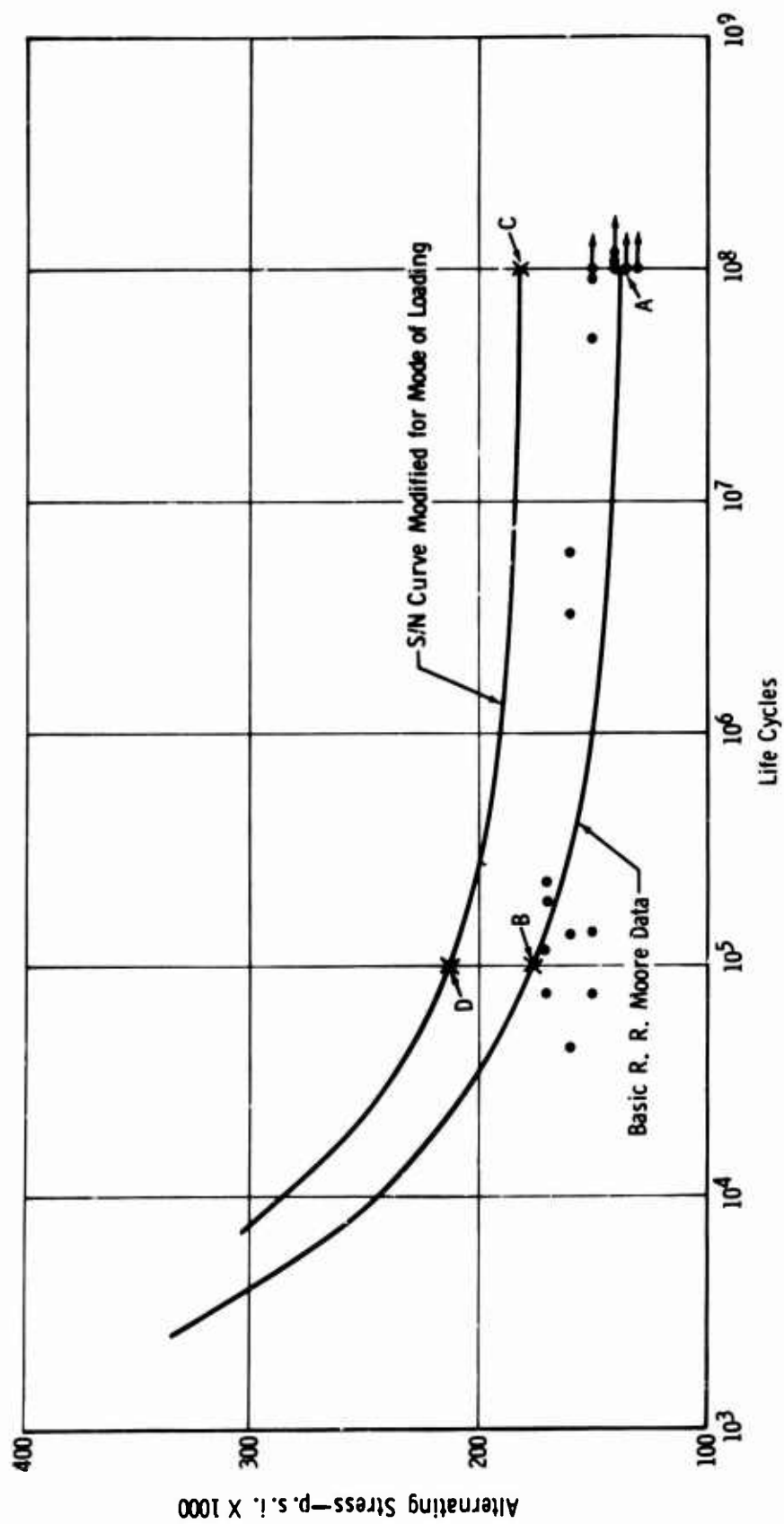
Figure 89. R. R. Moore Fatigue Test Data.

116
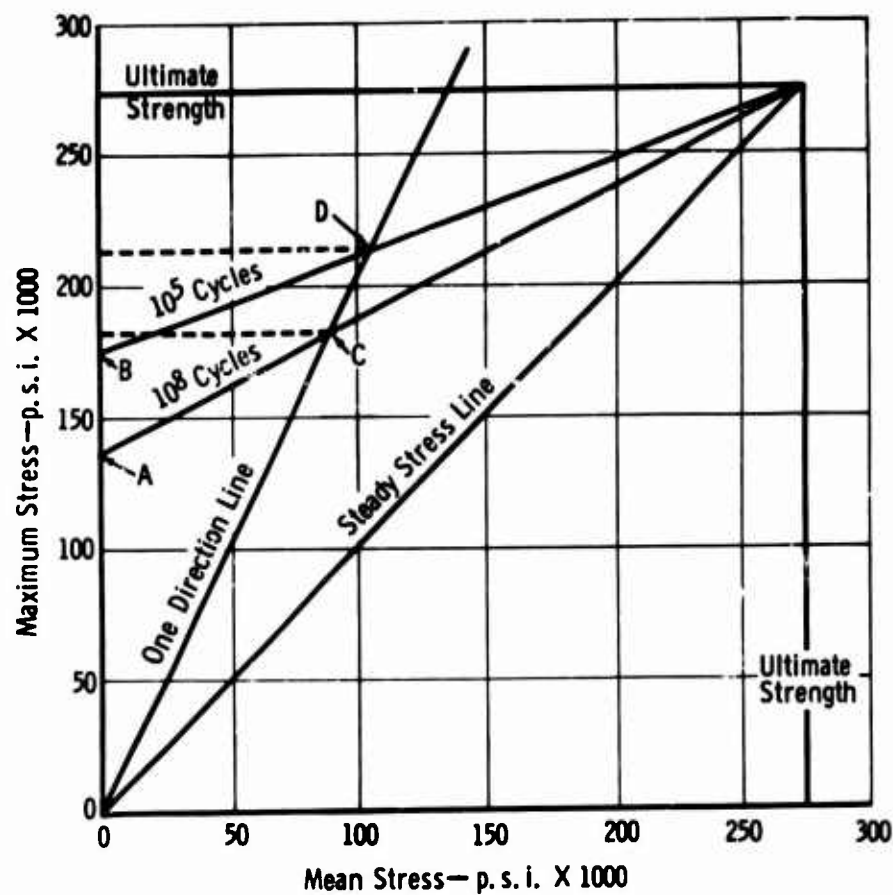
Figure 90. Modified Goodman Diagram.

Size Effect

R. R. Moore standard specimens are 0.250-inch-diameter bars. Generally, for bending, the endurance strength tends to decrease as size increases. To relate the size effect factor to carburized gears, it is recommended that the factor be "one." The literature indicates that the decrease of endurance strength for size is approximately 2 percent for carburized material; however, this effect has not been completely tested.

Surface Effect

Usually R. R. Moore specimens are polished. For this analysis, however, the R. R. Moore specimens were ground to the same surface finish as the gear roots; thus, the surface effect factor is "one." R. R. Moore data from polished samples must be reduced 10 percent.

Stress Concentration

R. R. Moore specimens are considered to have no stress concentration. Most current gear tooth bending stress calculation methods incorporate a stress concentration term based on tooth geometry. Therefore, no further consideration of stress concentration is required.

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### Reliability

Both R. R. Moore and fatigue test data have been analyzed based on mean endurance strength (50 percent failures) for comparison. Depending on the application, any confidence level may be selected for the gear design.

### Surface Treatment

The R. R. Moore samples in this program were carburized, shot peened, and black oxided to the same specifications as the gears. Thus, the surface treatment factor is "one."

All of the aforementioned factors except stress concentration, size effect, and mode of loading are considered as one for this analysis. Thus, the modified R. R. Moore data as plotted on the S/N curve of Figure 89 are comparable (within 2 percent) to a calculated stress that incorporates a stress concentration factor.

Figures 91, 92, 93, and 94 show the fatigue test data with respect to size and pressure angle plotted against AGMA stress. Superimposed on these curves is the endurance strength line from the modified R. R. Moore data developed previously. It is considered significant that close correlation is indicated for the AGMA method and the basic R. R. Moore data. A further comparison is made in Figures 95 and 96 by superimposing the R. R. Moore S/N curve on the protuberance hobbed and the full form ground data. A final comparison is made by averaging the fatigue test gear data and comparing with the R. R. Moore S/N curve. Figure 97 shows this comparison. It is apparent that extremely close correlation was demonstrated between the overall AGMA stress calculation for the gear fatigue tests and the basic strength as determined by the R. R. Moore data.

The endurance strengths previously listed in Tables XXII, XXIII, and XXIV are plotted in Figure 98 and are compared to the basic R. R. Moore data. It is apparent that the Lewis, Heywood, and Kelley-Pedersen methods do not approach the basic material strength. The Dolan-Broghamer and AGMA methods, which are very similar, do bracket the basic material strength line.

### DEVELOPMENT OF DESIGN VALUE

The S/N curve of Figure 97 was obtained from an average of all the fatigue test data. It represents a mean or 50-percent failure estimate of the test data. For design purposes, a much lower failure probability would normally be required. An endurance limit consistent with such a higher reliability was obtained as follows. If some of the differences among the derived endurance limits are attributed to geometric factors and combined into one group, a distributed quantity results. The group of endurance limits has an average value and some scatter or dispersion about this average. A meaningful statement of the form of this distribution is not possible because there are only 16 points. However, a plot of these points on normal probability paper (Figure 99), using the mean rank procedure, indicates that an assumption of normalcy is reasonable. Assuming normalcy, a lower tolerance value can be calculated for the endurance limit. The average, $\bar{X}$, and standard deviations of the distribution were calculated after deleting the endurance limit derived from configuration 3. The K factor for a one-sided tolerance limit was obtained from tables which can be found in standard statistical texts. This K factor for a proportion P = 0.99 and a probability of 0.80 is 3.212. The 1-percent endurance limit is then $\bar{X} - K_a$ or 182,000 - 3.212 (24,900) = 102,000 p.s.i. The
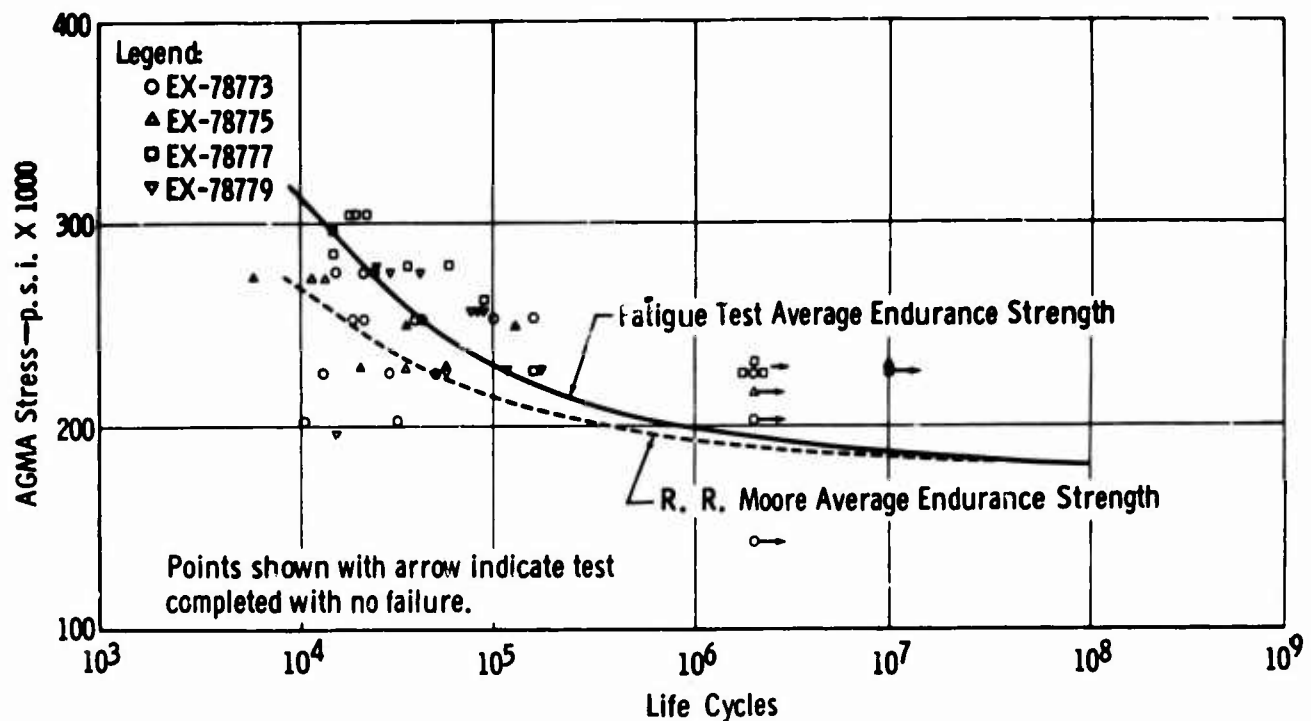
Figure 91.   AGMA Stress Fatigue Test Data
(Diametral Pitch = 12;  Pitch Diameter = 2 Inches;  Pressure Angle = 20 Degrees).
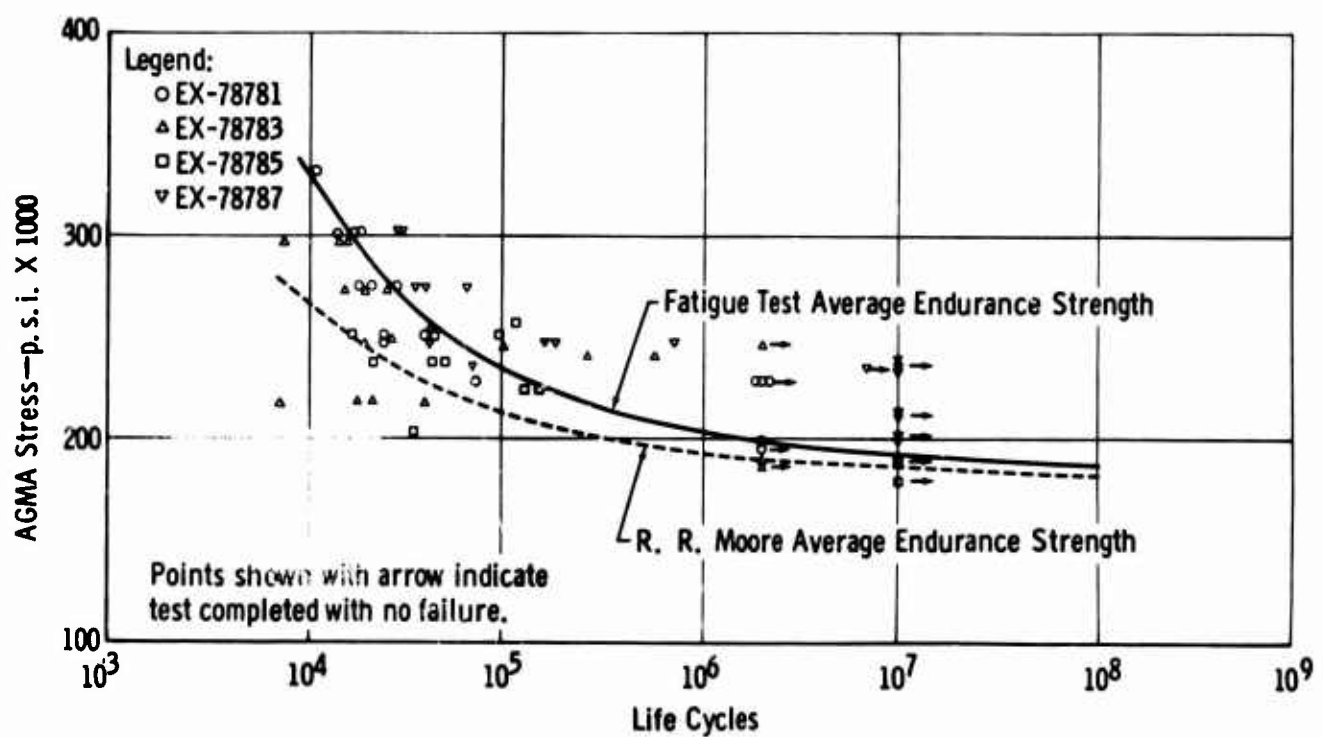


Figure 92.   AGMA Stress Fatigue Test Data
(Diametral Pitch = 12;  Pitch Diameter = 2 Inches;  Pressure Angle = 25 Degrees).

119
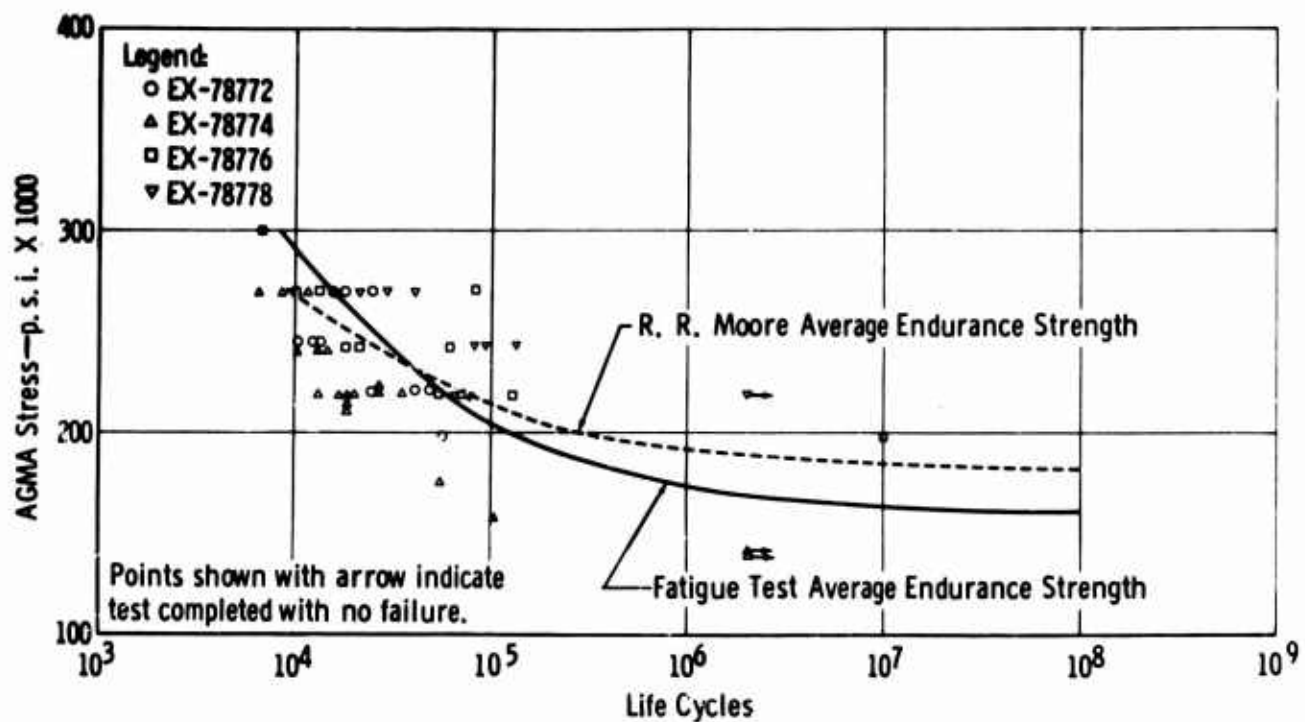
Figure 93. AGMA Stress Fatigue Test Data
(Diametral Pitch = 6; Pitch Diameter = 4 Inches; Pressure Angle = 20 Degrees).
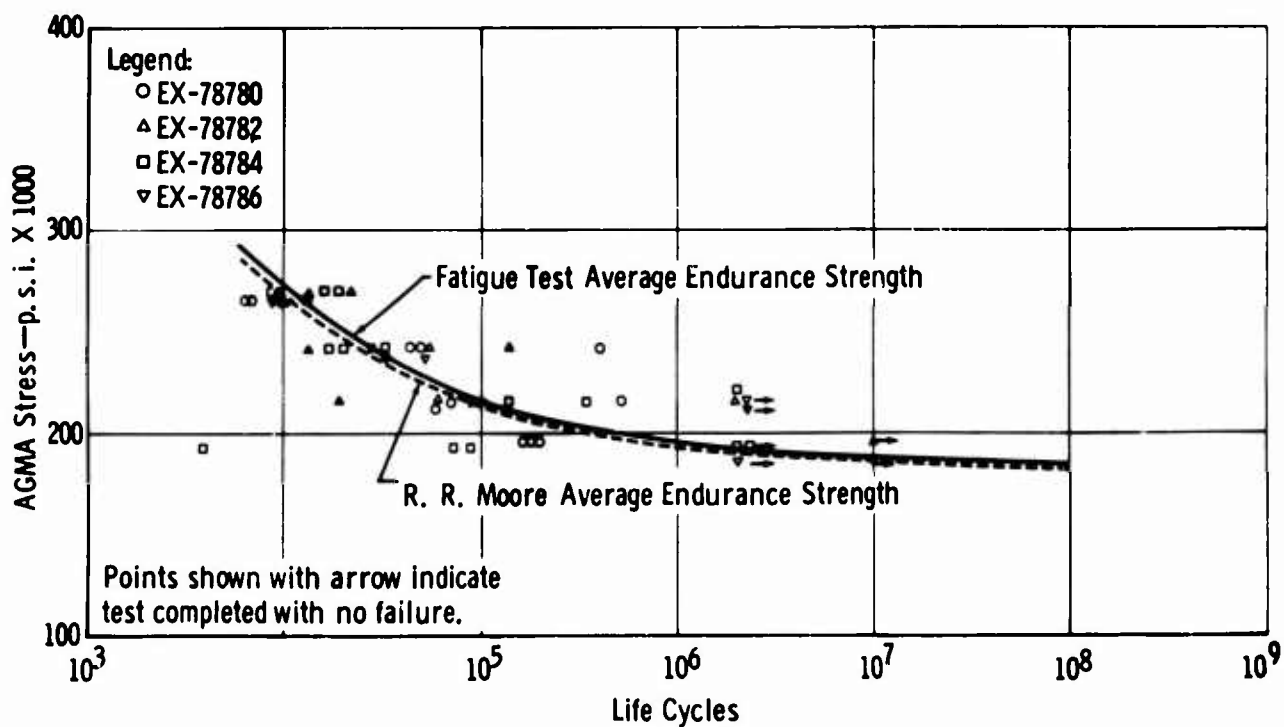


Figure 94. AGMA Stress Fatigue Test Data
(Diametral Pitch = 6; Pitch Diameter = 4 Inches; Pressure Angle = 25 Degrees).

120
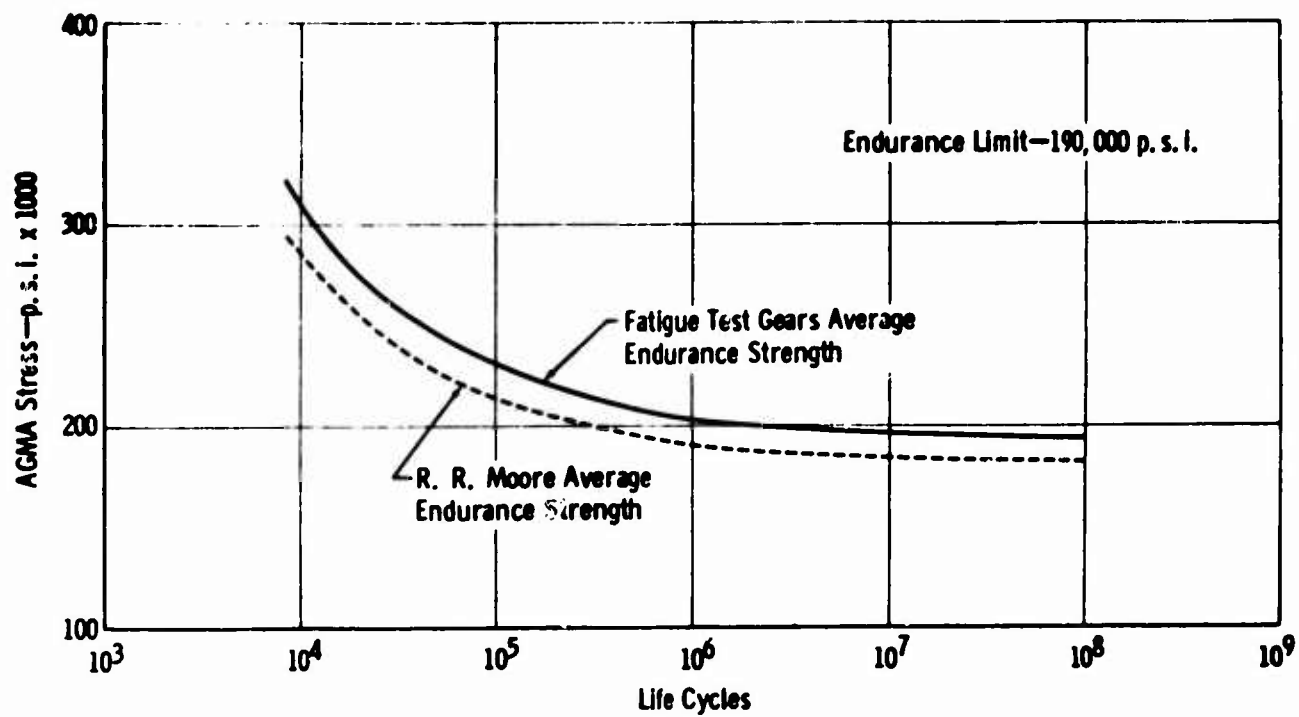
Figure 95.   S/N Diagram for Protuberant Fillet.
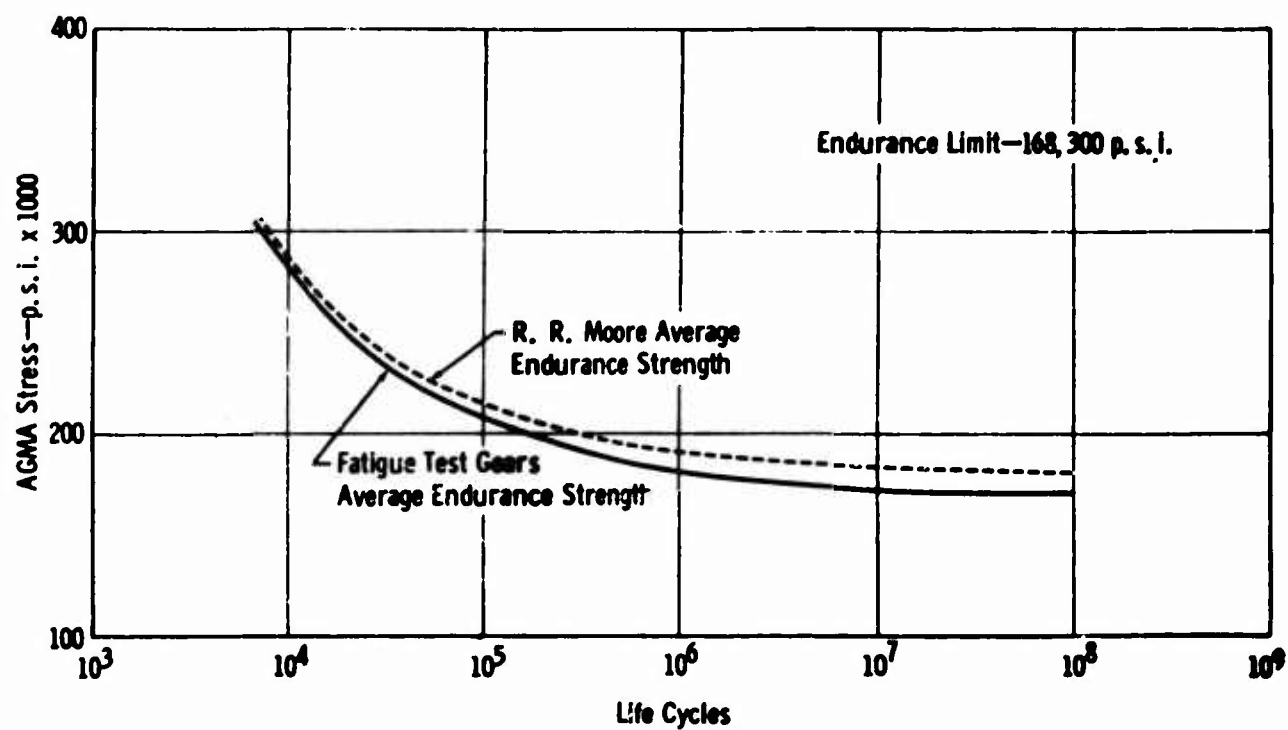


Figure 96.   S/N Diagram for Full Form Ground Fillet.

121
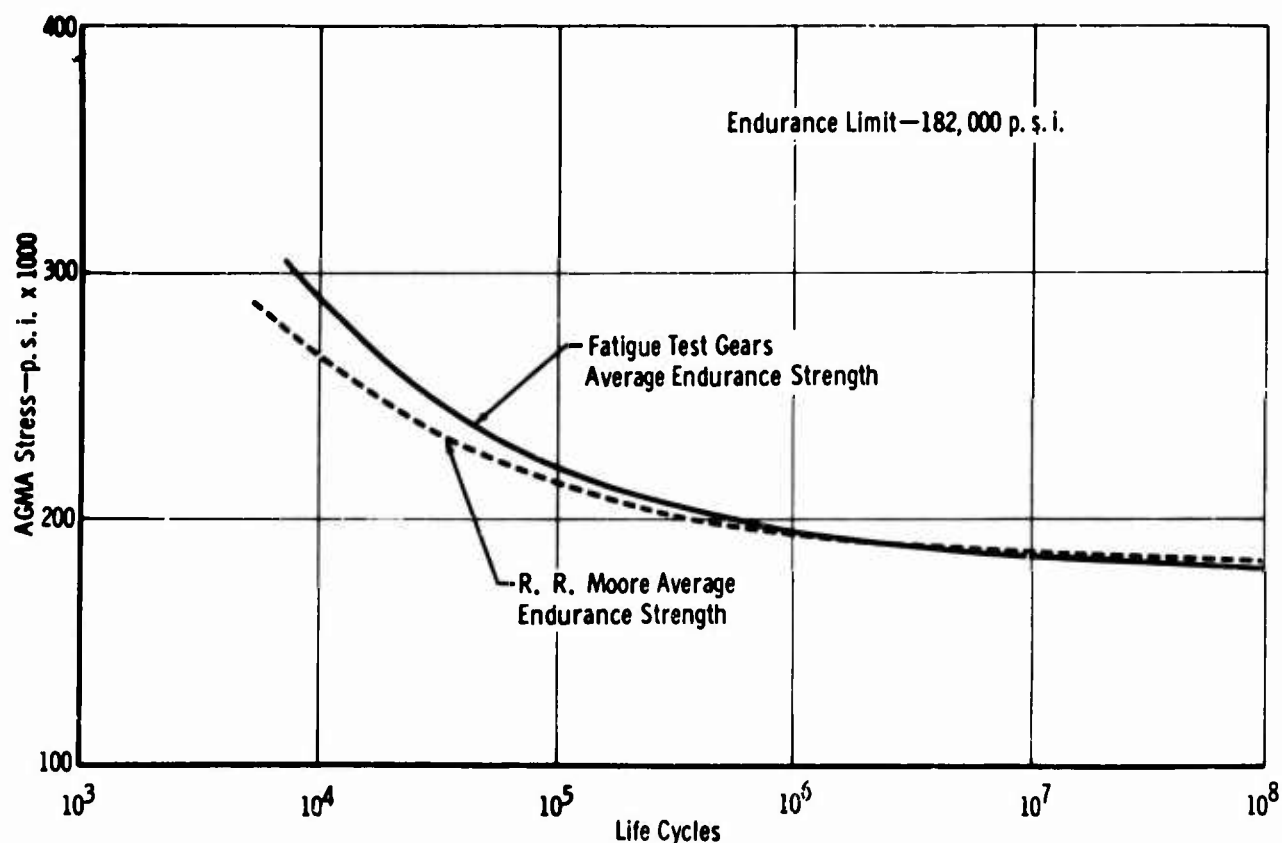
Figure 97. Average Fatigue Endurance Strengths Compared With R. R. Moore Data.

probability statement then is: "There is 95 percent probability (confidence) that at least 99 percent of the endurance limits of gears will be greater than 102,000 p. s. i.". Thus, a fatigue reliability factor of approximately 182,000/102,000 = 1.78 is indicated.

The S/N curve representing the overall average and a tolerance representing 1-percent failure are shown in Figure 100. Using the 1-percent line as a design value, it is estimated that 1 percent of the gear teeth will experience failure in bending. This statement is only an approximation, being restricted by the range of variables investigated, the significant effect of some of the geometric factors, and the limited knowledge of relating failure analysis of a single tooth to the probability of failure of one or more teeth on a gear.

LITERATURE COMPARISON

A comparison of the data with the literature indicates good correlation. Figures 101 through 104 show a comparison of the fatigue test points with the data published in reference 54. The data in the paper have been reduced to AGMA stress for comparison with the fatigue test data. In general the scatter is similar, with some fatigue points showing early failures.

Additional comparison was made with AGMA Proposed Standard 411.02, which specifies allowable endurance life values with load and stress distribution factors. This comparison is shown in Figure 105. Table XXV summarizes these data for AGMA, R. R. Moore, and the fatigue test gears. There is close correlation of the gear test data

122
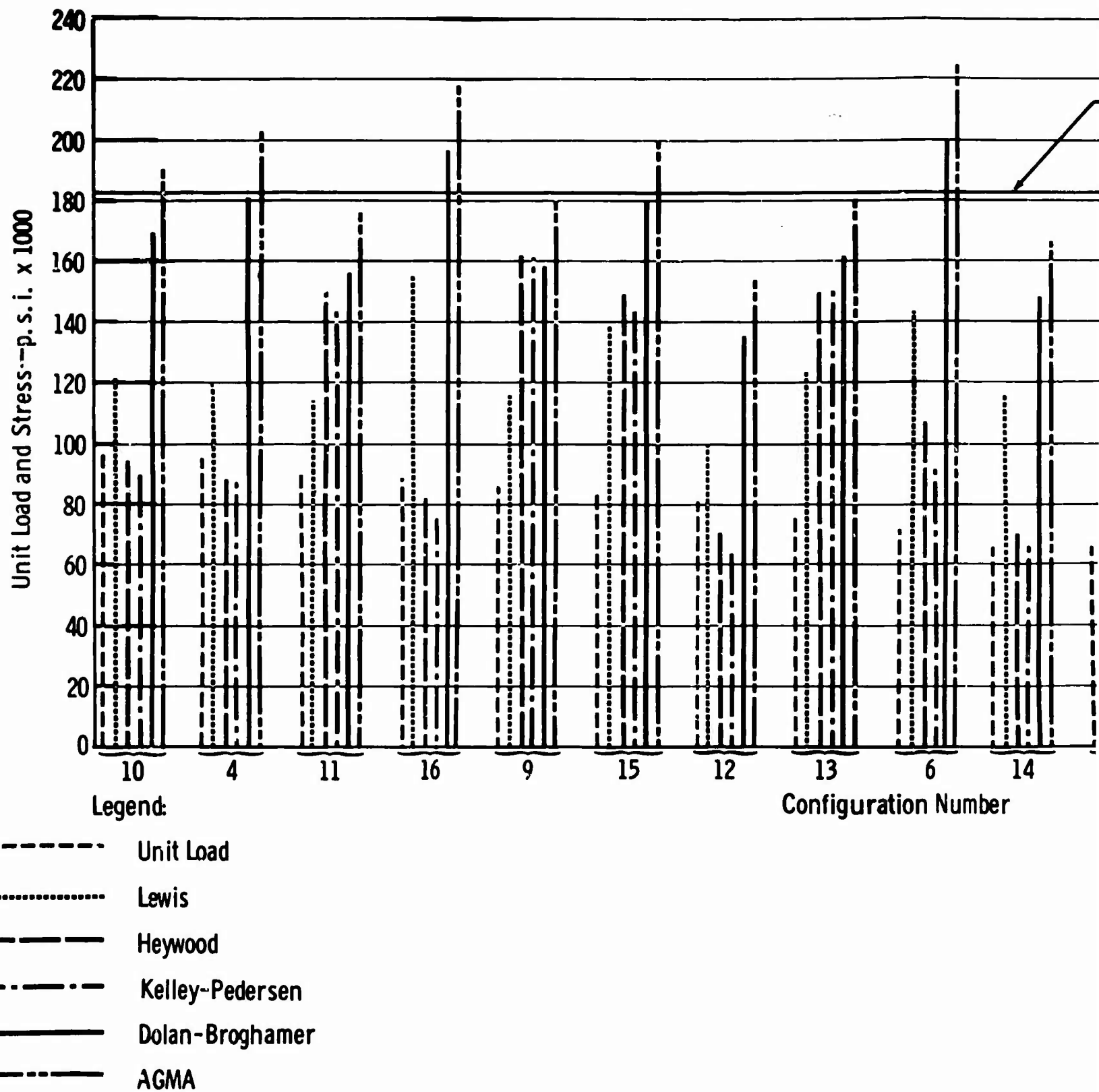
Figure 98. Methods of Calculating Stress for Endurance Strength Based on
Fatigue Test Gears Compared With R. R. Moore Endurance Strength.

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A

R. R. Moore Average
Endurance Strength

| 6 | 14 | 7 | 2 | 1 | 8 | 5 | 3 |

juration Number

Spread
of
Calculated
Stress
as
Endurance
Strength
for
Test
Gears
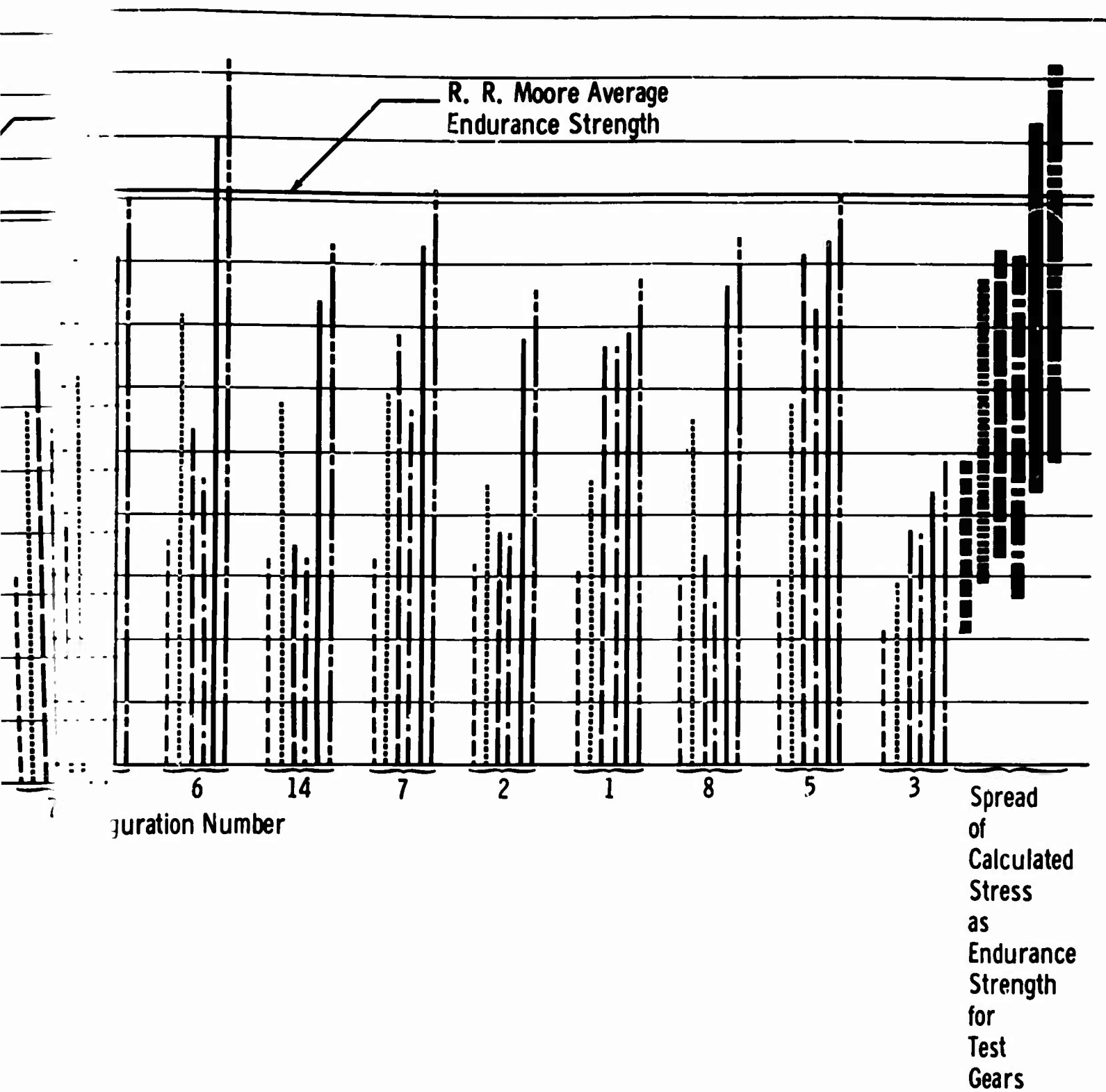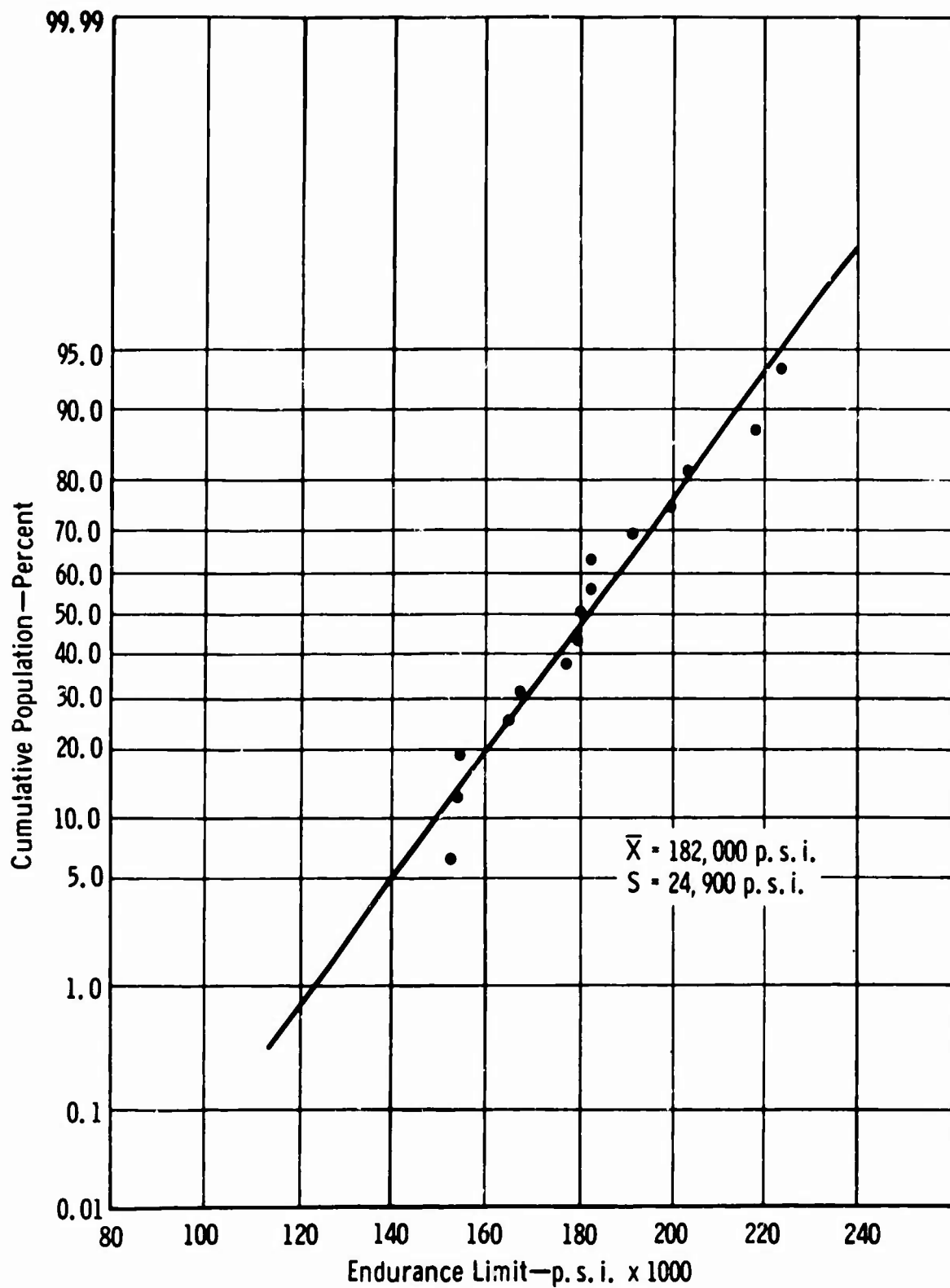
Figure 99. Distribution of Endurance Limits.
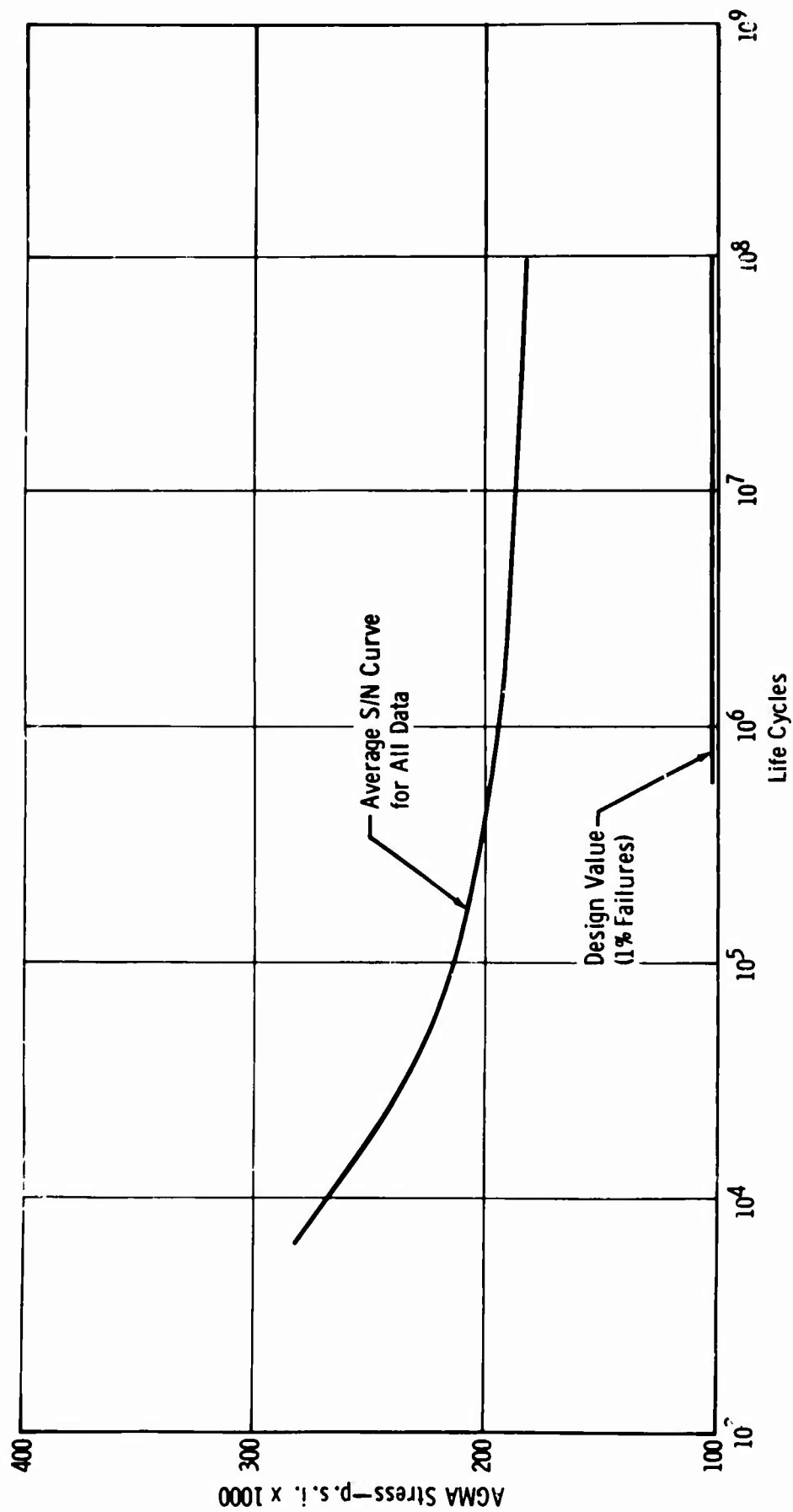
Figure 100. AGMA Average S/N Curve and Design Value.

endurance strengths for $10^7$ cycle life with the basic R. R. Moore data. The selection of the load and stress distribution factors for the fatigue test gears was based on the dynamic tests (Figure 109) for a gear at 16,000 feet/minute pitch-line velocity. It is obvious that selection of the various load and stress distribution factors may change the calculated stress appreciably.

## EVALUATION OF DYNAMIC EFFECTS

### Centrifugal Stress

Centrifugal stress consists of two major parts—hoop stress and centrifugal force stress. The hoop stress is a circumferential tensile stress at the root diameter caused by the tendency of the rim to expand from centrifugal force. The centrifugal force stress is a radial tensile stress caused by the centrifugal force exerted by the gear tooth.

The measured centrifugal stress was found to be much higher than the calculated stress caused by centrifugal forces on the gear teeth. However, the measured stress was found to coincide closely with the calculated hoop stress. This was true for both the root and the active profile positions. This suggested that the hoop stress spread onto the active profile of the gear tooth. Figure 106 shows a comparison of calculated centrifugal force stress, calculated hoop stress, and measured centrifugal stress. The measured stress was found to be 75 percent of the calculated hoop stress.
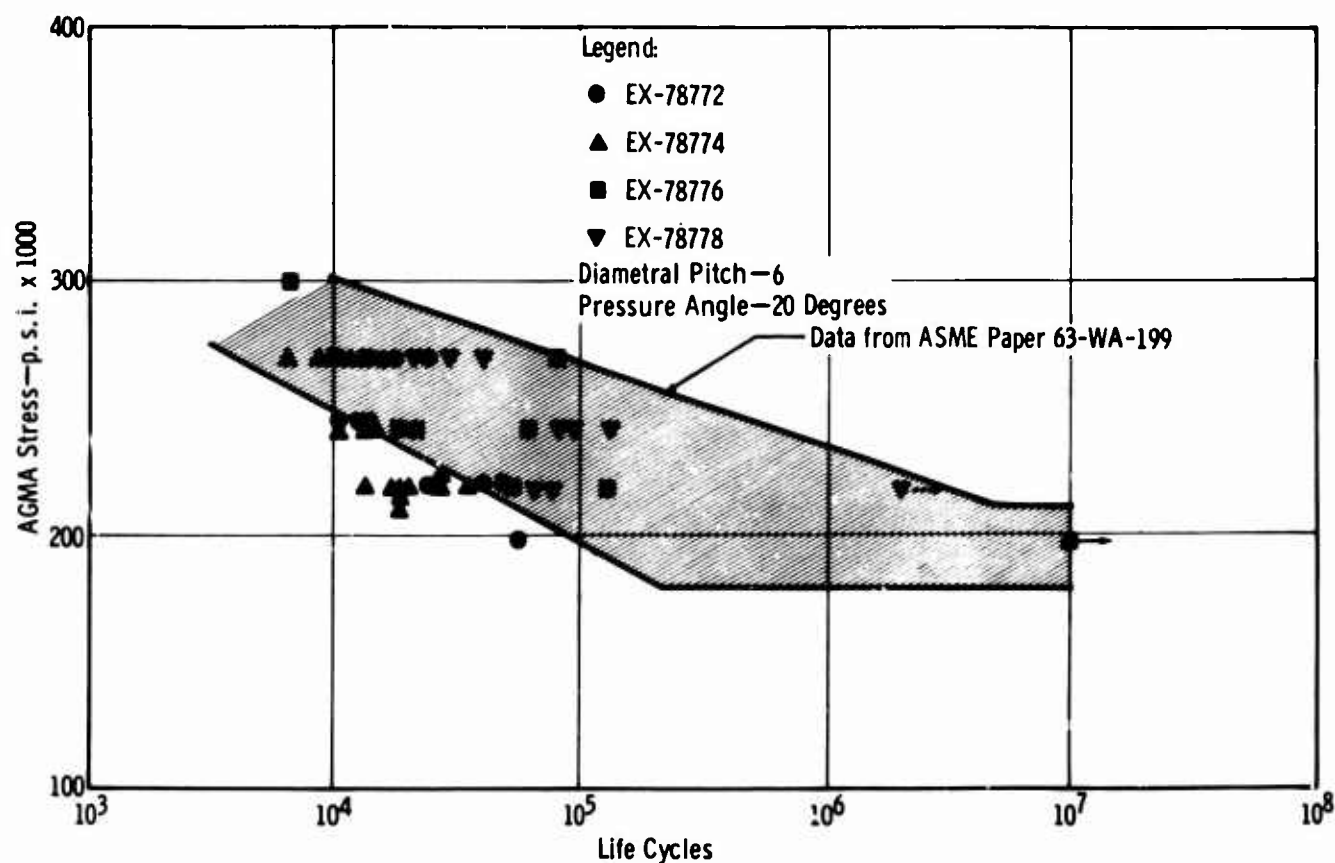


Figure 101. Comparison of Test Data With ASME Paper 63-WA-199 (Reference 54).

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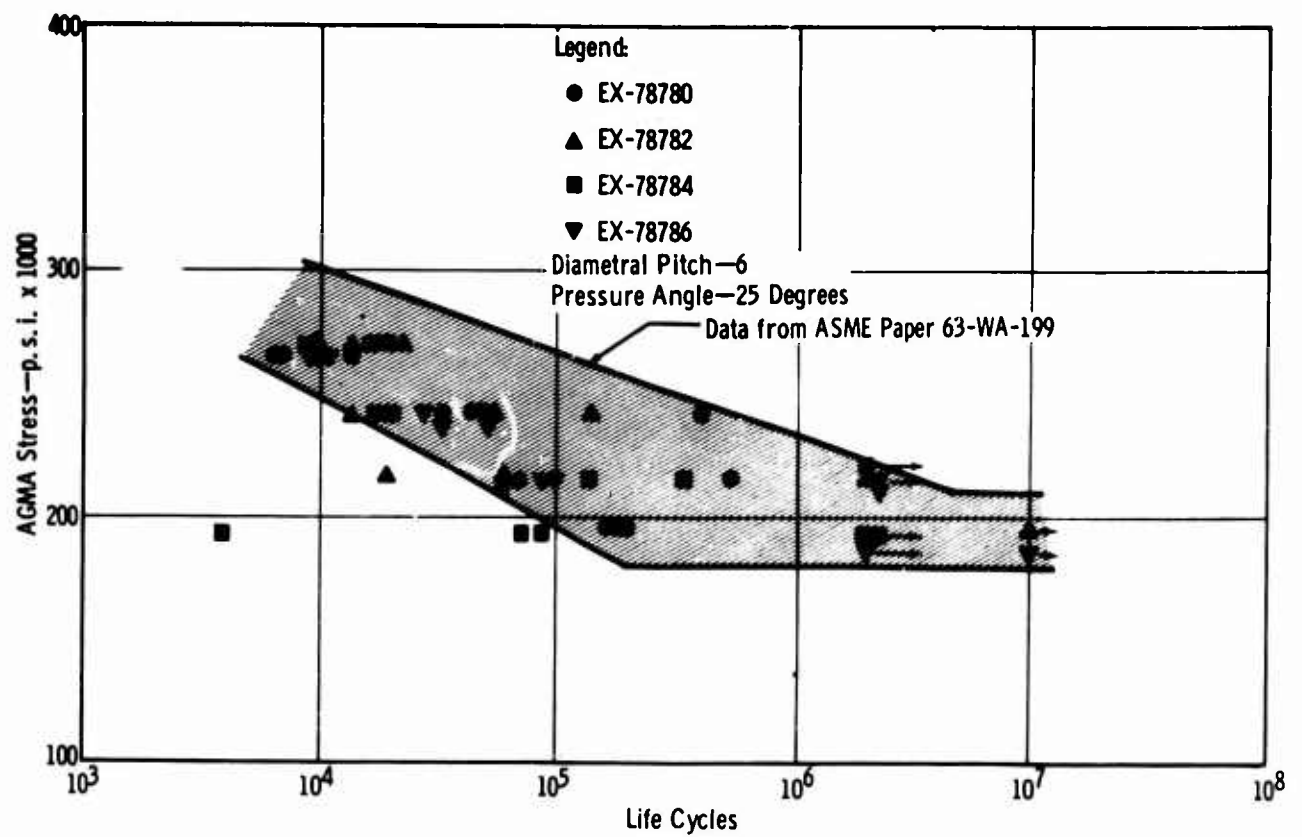
Figure 102. Comparison of Test Data With ASME Paper 63-WA-199 (Reference 54).
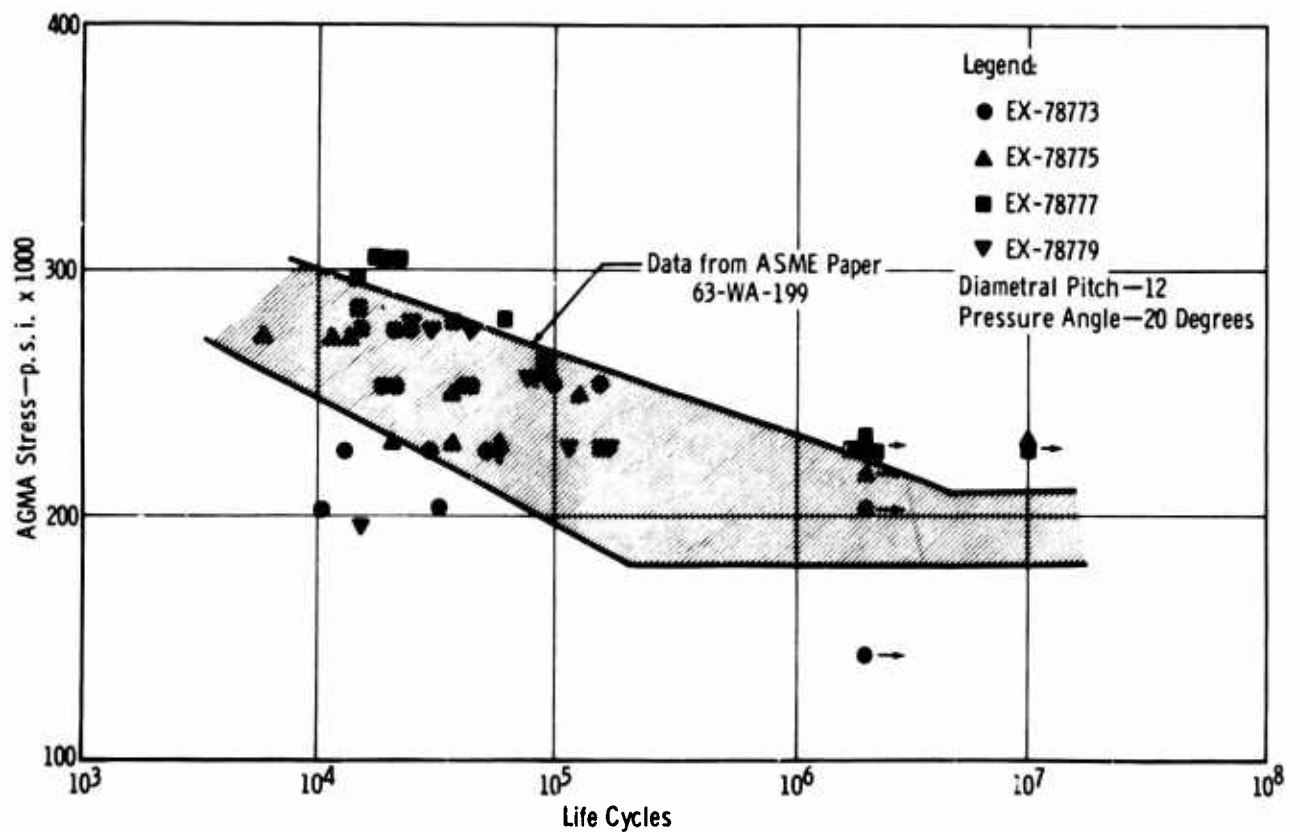


Figure 103. Comparison of Test Data With ASME Paper 63-WA-199 (Reference 54).

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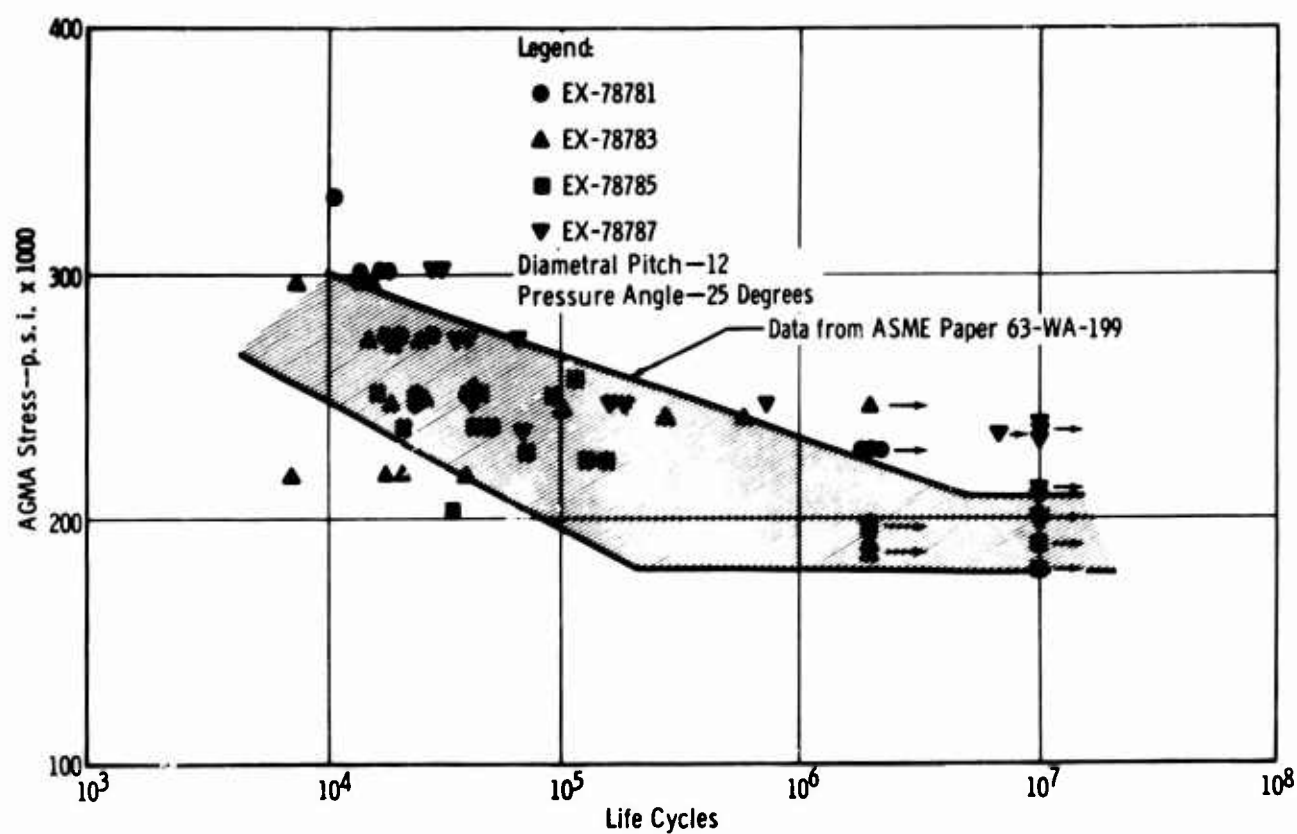
Figure 104.   Comparison of Test Data With ASME Paper 63-WA-199 (Reference 54).
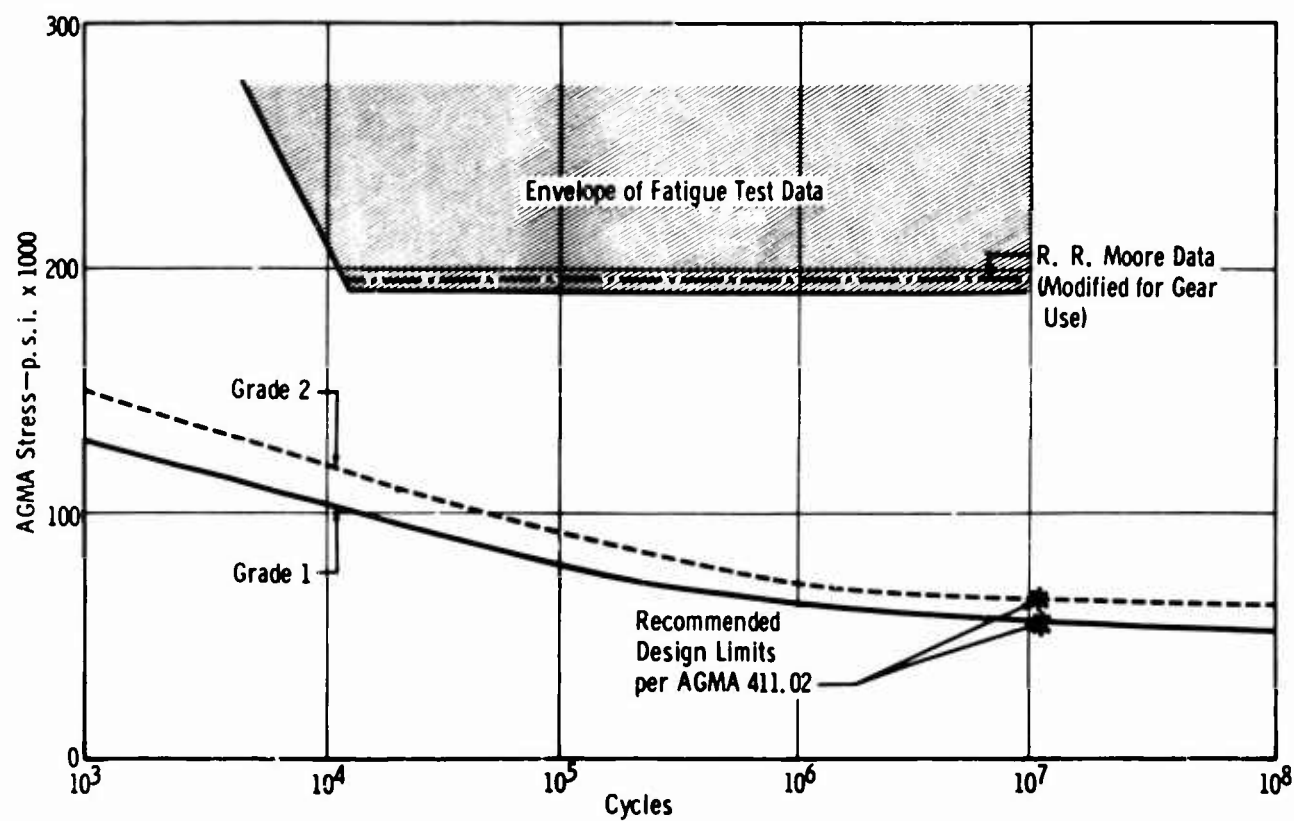


Figure 105.   Comparison of Test Data With AGMA Standard 411.02 Design Limits.

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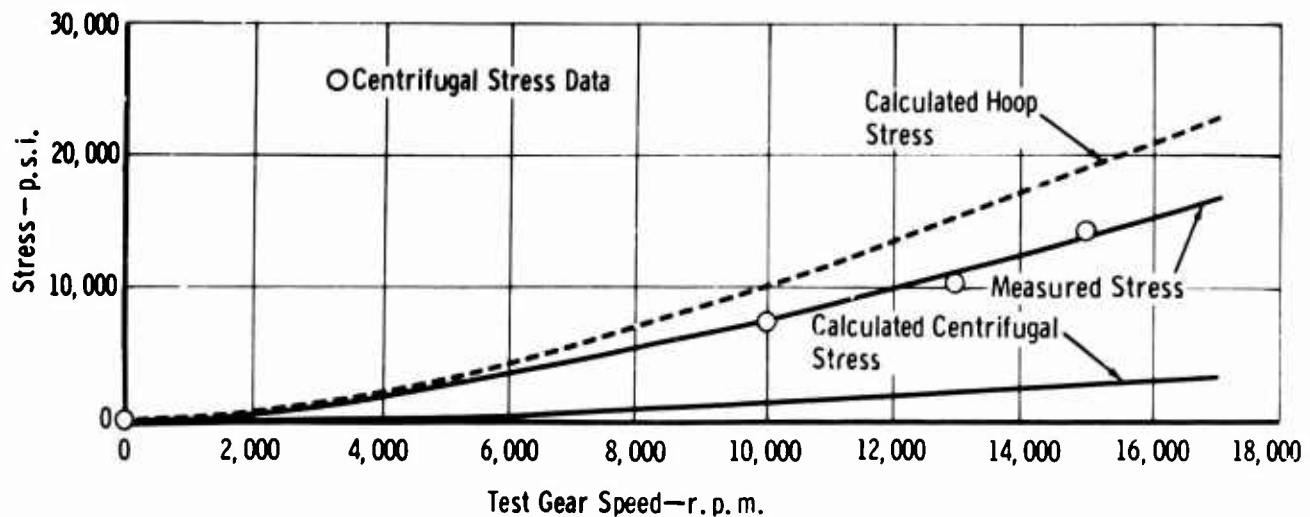
Figure 106. Comparison of Calculated and Measured Gear Stresses.

No detailed study was made of the possible effect of various gear tooth geometries and/or rim proportions on centrifugal stress at the weakest section. The similarity of the hoop stress and centrifugal force formula, both of which vary with the square of the speed, and the similarity of normal gear tooth geometry (unit diametral pitch rule) suggest that the observed proportional values should remain essentially constant. Design use of the calculated hoop stress should therefore be conservative.

Hoop stress, $S_h$, can be calculated by the following equation:

$$S_h = P \frac{V^2}{g}$$

where

$V$ = velocity at rim, inches/second
$P$ = material density, pounds/cubic inch
$g$ = gravitational acceleration constant, 386 inches/second squared

Since the stress was desired at the root diameter, the equation may be expressed as:

$$S_h = \frac{N}{60g} P D_r = 0.000136 \, PND_r$$

where

$N$ = rotational speed, r.p.m.
$D_r$ = root diameter, inches
$P$ = material density, pounds/cubic inch

Since the centrifugal stress is at a constant level (at constant speed), use of a modified Goodman diagram was required to permit combining with the alternating bending stress from the normal tooth load. See Figure 107. The S/N curve developed from the fatigue test program (Figure 97) was used at the zero centrifugal stress ordinate to construct the modified Goodman diagram. The Goodman diagram may be used to determine the endurance strength required for the bending stress calculation given a desired life, speed, and gear size.

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## TABLE XXV
### COMPARISON OF FATIGUE TEST DATA

| | Load | | | Stress Distribution | | | Endurance Strength from Test Data (p.s.i.) | Allowable Stress for $10^7$ Cycle Life (p.s.i.) |
| | $K_o$ (Overload) | $K_v$ (Dynamic) | $K_s$ (Size) | $K_m$ (Distribution) | J (Geometry) | Centrifugal Factor | | |
|---|---|---|---|---|---|---|---|---|
| AGMA (for typical aircraft) | Section 9* Table 4 | Section 8* Figure 5 | Section 7* | Section 6* Table 3 | Section 5* | — | — | 65,000 Grade 2** |
| | 1.0 | 1.0 | 1.0 | 1.0 | 0.33 | — | — | 55,000 Grade 1** |
| ASME 63-WA-199 (reference 54) | 1.0 | 1.0 | 1.0 | 1.0 | 0.33 | — | 180,000 | 60,000 |
| R. R. Moore*** | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | — | 195,000 | — |
| Fatigue Tests | 1.0 | 1.0 | 1.0 | 1.0 | 0.425 | — | 182,000 | 102,000 (1% Failures) |
| Dynamic Tests | 1.0 | 1.3† | 1.0 | 1.0 | 0.425‡ | 1.1§ | 182,000 | 71,000 (1% Failures) |

*AGMA Standard 220.02 (Appendix VI herein).

**AGMA Standard 411.02.

***Corrected data presented in the subsection titled Basic Material Strength.

† Dynamic factor for 16,000 feet/minute per Figure 109.

‡ Stress concentration factor—average of test gears based on Lewis and measured strain gage data = 2.42.

§ Centrifugal factor at 16,000 feet/minute per Figure 106.
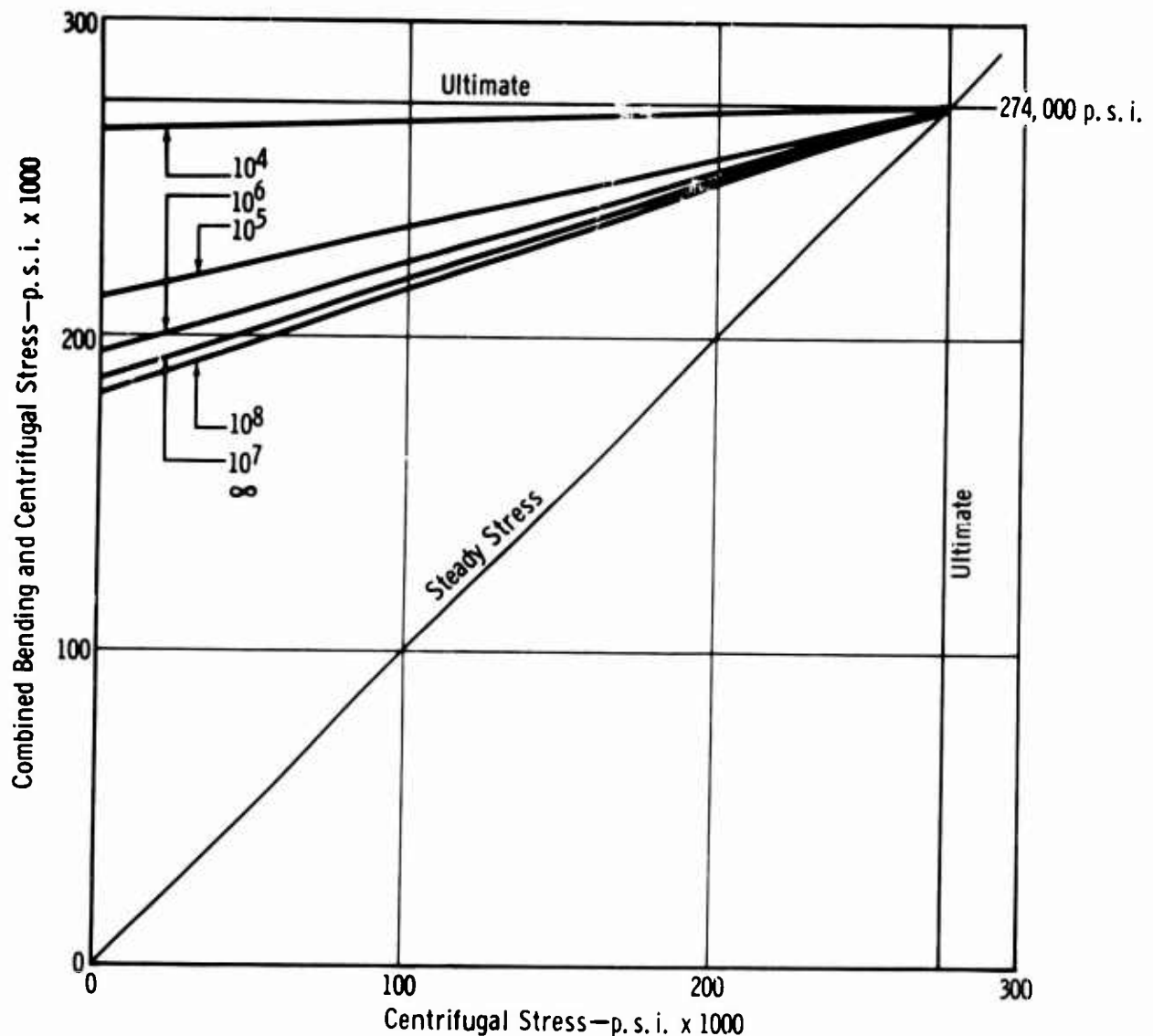
Figure 107. Modified Goodman Diagram Combining Centrifugal and Bending Stresses.

For example, the dynamic test gear when operating at 16,000 r.p.m. has a calculated hoop stress of 20,000 p.s.i. For $10^7$ cycle life, a bending stress of 175,000 p.s.i. would be permitted based on the modified Goodman diagram. Based on direct addition of the centrifugal and bending stress (an improper procedure), the S/N curve would permit only 162,000-p.s.i. bending stress. Also, this gear, if designed for $10^7$ y le life without considering centrifugal stress, would actually have a mean life expectar y of slightly less than $10^5$ cycles or only 1 percent of that anticipated. To calculate a more comprehensive gear tooth bending stress under high-speed operating conditions, the hoop stress must be combined with bending stress by use of the modified Goodman diagram.

Dynamic Stress

Figure 108 is a plot of the peak dynamic stress versus r.p.m. The strain readings were converted to stress and plotted against gear r.p.m. for three load conditions—380,

570, and 766 pounds (1000, 1400, and 2000 pounds/inch of face width). The curves represent the best fit square curve above the static base line; thus, the amount of increase above the static stress level is equal to the square of the ratio of the speed. The static stress level is the measured stress at zero r.p.m. for pure tangential load. It was felt that a square curve would be the most desirable, since the dynamic effect could be related to kinetic energy which involves velocity squared. Again, the measured dynamic stress does not include any constant centrifugal stress.

Figure 109 shows a dynamic stress correction factor derived from the curves in Figure 108.

Figure 110 is a comparison of the dynamic factor as previously described with the one given in AGMA Standards 220.02 (Appendix VI). Curves 1, 2, and 3 represent various grades of gear quality with 1 being the highest quality gear. The propeller brake gear used in testing would be defined as a grade 1 gear. The two curves agree within 8 percent at 8000 feet/minute. Also, the AGMA data do not exceed 8000 feet/minute.

Although the dynamic data presented are very limited, they do indicate trends for high speed, lightweight gearing. It is recommended, therefore, that the curve of Figure 109 be used as a design factor for applications above 8000 feet/minute. Below this speed, a factor of one should be satisfactory for close-tolerance aircraft applications.
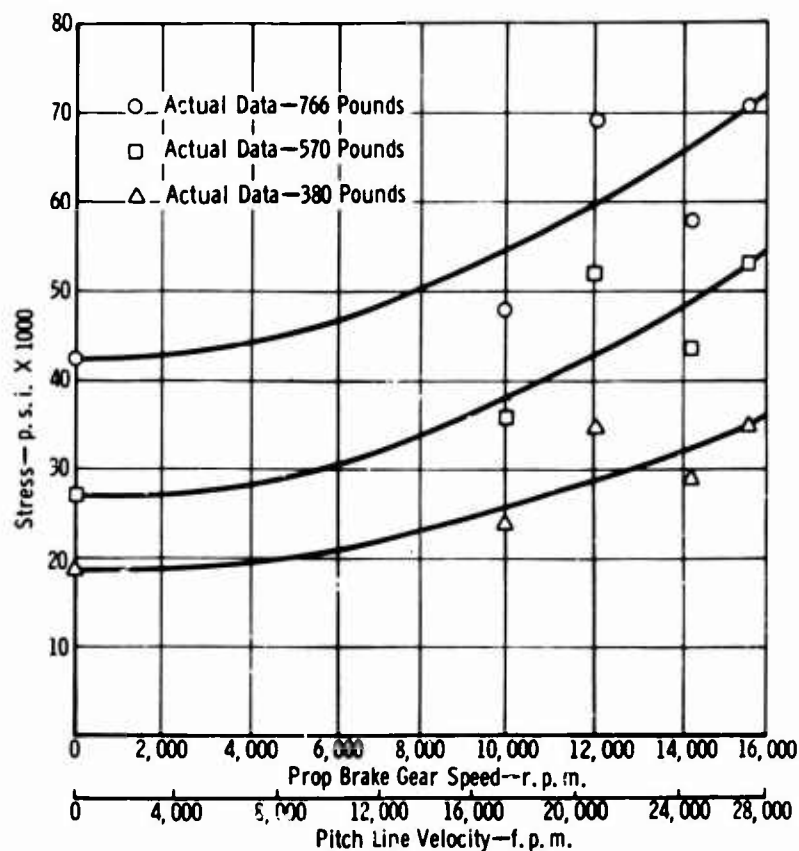


Figure 108. Graph Showing Peak Dynamic Stresses During Testing.

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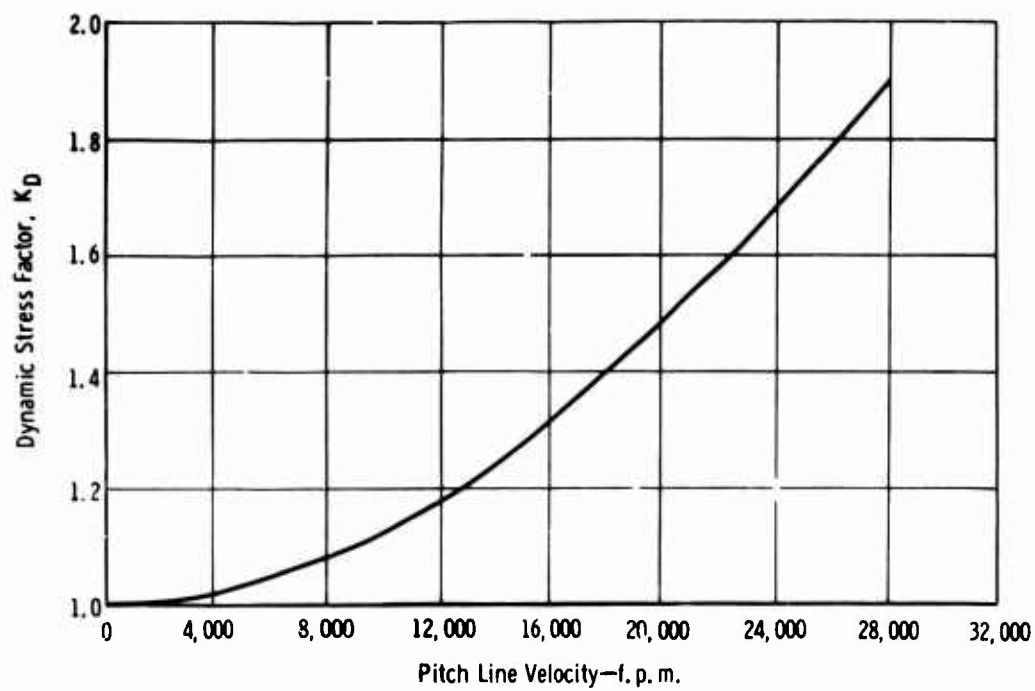
Figure 109. Dynamic Stress Factor as a Function of Pitch Line Velocity.
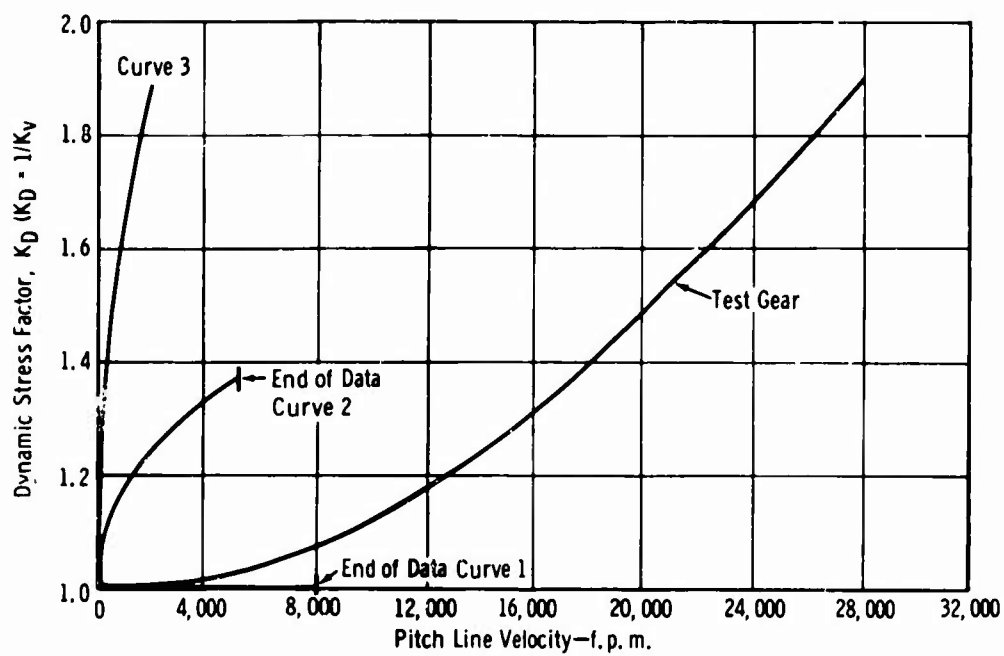


Figure 110. Comparison of Dynamic Stress Factors.

## ESTABLISHMENT OF COMPUTER PROGRAM

Analysis of the fatigue test data indicates that the AGMA formula is the most accurate for predicting ranking, produces the least variation in calculated endurance limits, best matches experimentally measured stresses, and accommodates the geometric variables with the least difference of significant values. The AGMA formula was selected, therefore, for use in the computer program. The AGMA formula also is a well known method—it is required by some Government specifications (reference 47).

The Lewis gear tooth geometry form factor values (Y), as calculated by the computer, should be more accurate than values normally obtained by graphical layouts. The point of tangency between the inscribed parabola and the generated trochoidal root fillet as well as the trochoidal root fillet contour can be established with precision.

A dynamic factor is an input item of the computer program. The dynamic factor for a given application may be obtained from existing AGMA curves, the curve presented in Figure 109, literature sources, or from direct "in-house" measurements.

Hoop stress is calculated in the program and combined with the AGMA calculated bending stress based on the modified Goodman diagram. A mathematical expression for the combined stress is:

$$S_c = US - \frac{US \left[ US - (S_h + S_t) \right]}{US - S_h}$$

where

$S_c$ = combined stress, p.s.i.
$S_h$ = hoop stress, p.s.i. (reference page 130)
$S_t$ = tensile stress (AGMA), p.s.i. (reference page 10)
US = ultimate strength of the material, p.s.i.

Life cycles are then determined from the combined stress and the S/N curve based on R. R. Moore rotating beam tests of the gear material. The life may be modified further by the AGMA temperature factor and reliability factor (factor of safety) as indicated by the expression:

$$L \propto S_c K_T K_R$$

where

L = life in cycles
$S_c$ = combined stress, p.s.i.
$K_T$ = AGMA temperature factor (reference page 11)
$K_R$ = AGMA factor of safety (reference page 11)

The term $\propto$ indicates the S/N curve stress-to-life cycle relationship.

Both AGMA bending stress and the combined bending and hoop stresses are printed out. Life is printed out if it is in the finite life area of the modified Goodman diagram; otherwise, an infinite life or an excessive stress note is printed.

Considerable effort was expended in graphical analysis of the Lewis gear tooth form factor Y and its relationship to the Dolan-Broghamer stress concentration factor $K_f$.

It is expected that strength and stress concentration factors should be geometrically related. Gear sets with the following range of parameters were computed and plotted:

- Pressure angle—14.5, 20, and 25 degrees
- Number of teeth in pinion—12 through 52
- Gear ratio—1.0 through 10.0
- Hob tip radius—100, 75, 50, and 25 percent of maximum possible
- Dedendum factor—1.157, 1.2, 1.3, and 1.4
- Tooth thickness at pitch diameter—100, 90, and 110 percent of half of the circular pitch

The parametric plots were not smooth, overlapping curves as expected. The original Dolan-Broghamer data (from reference 16) were therefore analyzed. Computer-determined dimensions (h, t, and $\theta_L$) for the given gear teeth do not coincide with the dimensions for the plastic models as tabulated in reference 16. The computer values plot as smooth curves while the original data do not; this indicates that the error is most likely that which is inherent with the drafting layout procedure. Computed $K_f$ values based on corrected geometry and observed stresses produce data which vary by ± 11 percent from that computed by the formula as indicated in Table XXVI.

Work to generate a formula to duplicate the corrected stress concentration factors obtained has not been completed.

The Dolan-Broghamer photoelastic data were obtained from models having pressure angles of 14.5 and 20 degrees, diametral pitch of 2, and a dedendum factor of 1.157. Graphical analyses should be used with the new stress concentration formula to determine the validity of the extrapolation if $K_f$ values throughout the range of gear tooth geometric variables, as previously investigated. Similar analysis of additional photoelastic data (such as from Kelley-Pedersen work) would be valuable for correlation.

A new stress concentration factor, developed as described, would considerably enhance the correlation of the test data and would be a valuable modification to the AGMA formula and the computer program.

## TABLE XXVI
## COMPARISON OF STRESS CONCENTRATION FACTORS

| Model Number | $K_f$ (Dolan-Broghamer) | $K_f$ (AGMA) | $K_f$ (Calculated) | $K_f$(Calculated)/$K_f$(AGMA) |
|---|---|---|---|---|
| 6-1 | 1.53 | 1.511591 | 1.500636 | 0.99272 |
| 6-2 | 1.65 | 1.647910 | 1.733851 | 1.05218 |
| 6-3 | 1.82 | 1.832482 | 1.876810 | 1.02417 |
| 6-4 | 2.18 | 2.097727 | 2.117530 | 1.00943 |
| 6-5 | 1.56 | 1.558408 | 1.638576 | 1.05146 |
| 6-6 | 1.68 | 1.694056 | 1.817664 | 1.07295 |
| 6-7 | 1.86 | 1.877288 | 1.959995 | 1.04405 |
| 6-8 | 2.10 | 2.141456 | 2.287704 | 1.06826 |
| 6-9 | 1.68 | 1.644061 | 1.826440 | 1.11088 |
| 6-10 | 1.76 | 1.783399 | 1.936391 | 1.08579 |
| 6-11 | 1.94 | 1.970920 | 2.057944 | 1.04414 |
| 6-12 | 2.21 | 2.241207 | 2.314211 | 1.03257 |

TABLE XXVI (CONT)

| Model Number | $K_f$ (Dolan-Broghamer) | $K_f$ (AGMA) | $K_f$ (Calculated) | $K_f$(Calculated)/$K_f$(AGMA) |
|---|---|---|---|---|
| 7-1 | 1.57 | 1.588621 | 1.589230 | 1.00037 |
| 7-2 | 1.68 | 1.735900 | 1.746881 | 1.00633 |
| 7-3 | 1.93 | 1.936614 | 1.882616 | 0.97211 |
| 7-4 | 2.37 | 2.228237 | 2.168161 | 0.97307 |
| 7-5 | 1.69 | 1.664860 | 1.788942 | 1.07747 |
| 7-6 | 1.86 | 1.810860 | 1.843543 | 1.01800 |
| 7-7 | 2.04 | 2.008900 | 1.986026 | 0.98860 |
| 7-8 | 2.30 | 2.297209 | 2.124755 | 0.92495 |
| 7-9 | 1.74 | 1.750553 | 1.938912 | 1.10756 |
| 7-10 | 1.90 | 1.899773 | 2.085223 | 1.09758 |
| 7-11 | 2.10 | 2.101321 | 2.215057 | 1.05420 |
| 7-12 | 2.40 | 2.394263 | 2.368038 | 0.98905 |
| 8-1 | 1.62 | 1.629011 | 1.687625 | 1.03597 |
| 8-2 | 1.74 | 1.782054 | 1.782343 | 1.00011 |
| 8-3 | 1.94 | 1.991574 | 1.913581 | 0.96083 |
| 8-4 | 2.25 | 2.298240 | 2.063709 | 0.89796 |
| 8-5 | 1.74 | 1.724950 | 1.883809 | 1.09205 |
| 8-6 | 1.86 | 1.876333 | 1.956013 | 1.04247 |
| 8-7 | 2.06 | 2.082457 | 2.165639 | 1.03990 |
| 8-8 | 2.31 | 2.384652 | 2.271327 | 0.95244 |

Notes: $K_f$ (Dolan-Broghamer) from reference 16 based on observed stress.

$K_f$ (AGMA) computed by formula from corrected geometry.

$K_f$ (Calculated) computed from corrected geometry and observed stress.

# CONCLUSIONS

The following conclusions are made from this study.

- The investigation of four geometric variables indicated that the endurance strength was significantly affected by changes in pitch diameter and pressure angle. These effects were in some instances greater than those predicted by bending stress calculations. The effects of fillet size and fillet configuration—full form or protuberance—were not significant with respect to the endurance strength of the configurations tested. Stress calculations did not accurately consider the fillet configuration.

- A basic material strength curve for carburized AMS-6265 was established by R. R. Moore specimens. This strength curve correlated very closely with the AGMA method of calculating stress.

- By averaging all fatigue test data points, a design S/N curve was established. For design purposes, a 1-percent failure endurance strength of 102,000 p.s.i. was also established.

- Of the five strength formulas investigated, the AGMA bending strength formula provides the most accurate method for assessment of spur gear tooth bending strength.

- The limited dynamic testing conducted indicated that a dynamic factor for light-weight aircraft gears should be considered for applications with a pitch line velocity over 8000 feet/minute.

- A centrifugal speed factor is necessary for high pitch line velocity applications.

- A modification is required to the Dolan-Broghamer stress concentration factor used in the AGMA formula to consider tooth geometry more accurately.

- The AGMA formula modified to incorporate a centrifugal speed, a high speed dynamic factor, and to use R. R. Moore material strength data will produce an accurate estimate of gear tooth bending stress and life. The dynamic fluctuating stress calculated by the AGMA formula, $S_t = \dfrac{W_t K_o}{K_v} \dfrac{P_d}{F} \dfrac{K_s K_m}{J}$, is combined with the steady centrifugal hoop stress formula, $S_n = \rho \dfrac{V^2}{g}$, to produce a combined stress, $S_c$, as follows:

$$S_c = US - \frac{US \left[ US - (S_h + S_t) \right]}{US - S_h}$$

The terms are defined on page 135. Life cycles may then be determined from an S/N curve based on R. R. Moore rotating beam tests of the gear material. The life may be modified further by the AGMA temperature and reliability factors as follows:

$$L \propto S_c K_T K_R$$

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# APPENDIX I

## FATIGUE TEST GEAR DRAWINGS

This appendix consists of the fatigue test gear drawings for the 16 configurations tested. These drawings are shown in Figures 111 through 126. The spur gear main accessory drive and propeller brake outer member are shown in Figures 127 and 128, respectively.

BLANK PAGE

BREAK T'° .005 MAX

.015 -.030 ROUND BREAK

.050 R MIN

ENLARGED VIEW OF GEAR PR(
SCALE NONE

TOOTH SPACE SHALL BE FULLY GROUND INCLU
DIA AND ADJACENT FILLETS AFTER HEAT
DISCONTINUITY SHALL OCCUR AT THE BLEND
RADIUS WITH THE ROOT DIA AND INVOLUTE
PEEN THE ENTIRE TOOTH SPACE PER EPS
EPS 18176 AFTER GRINDING, THEN HONE SURFAC
REMAINING SURFACES MAY BE PEENED PER EPS
SPECIFICALLY CONTROLED BY A √ SYMBOL. S
GRINDING TO BE UNIFORM ON PROFILE, FILLET
DIA. WITHIN .002. MEASURE AND RECORD FI
OVER PINS AND ROOT DIA. BEFORE AND AFTE

.502
.498

1.910
1.900
TYPICAL
12 PLACES
SIDE A

REMOVE 12 TEETH SPACED AS SHOWN
GEAR TEETH ARE TO FINISHED SIZE
SIDE B    ALL GEAR TOOTH INSPECTION REQU
ARE COMPLETED

1.5005 DIA
1.5000   A

CSK 90° TO 1.560 DIA
(BOTH SIDES)

ELECTRO CHEMICAL ETCH ALLISON PART NO.
AND LAST CHANGE LETTER, "SER" AND SERIAL
NO. HERE PER AS 478-7A1 OR 7A2

ELECTRO CHEMICAL ETCH POSITION NUMBERS
ON TEETH AS SHOWN. RECORD INVOLUTE PROFILE
AND LEAD CHECKS FOR SIDE A OF TEETH
1, 2, 3, AND 4; AND FOR SIDE B OF TEETH X1,
X2, X3, AND X4. RECORD POSITION NUMBERS
ON TOOTH TO TOOTH SPACING ERROR
CHECKS

GEAR TOOTH CONTROL

| INVOLUTE PROFILE TOLERANCE | | | |
|---|---|---|---|
| SIDE A | OD | SIDE B | |
| 0.5° 1.8 U | .0000 -.0002 | .008 | |
| 12.0° 26.8 U | | PD | SAME |
| 23.6° 52.6 U | .0000 -.0002 | MTCA | |
| 24.5° 54.7 U | | APD | |
| 33.3° 73.3 U | | BCD | |

SPACING TOLERANCE
A .0002      B .0002

LEAD TOLERANCE
A ±.0001    B ±.0001

FULLNESS TOLERANCE
A .0000      B .0001

MAXIMUM HOLLOW IN FORM .0001
NOTE      LIMIT = .0147 IN.

SPUR GEAR DATA
    6 PITCH 24 TEETH
    20° PRESSURE ANGLE
    DISTANCE OVER TWO .2880 DIA PINS = 4.3999 +.0000 -.0046
        ROOT DIA = 3.583 ±.02
        PITCH DIA = 4.0000
        OUTSIDE DIA = 4.3333 ±.0000
    ACTIVE PROFILE OUTSIDE 3.7984 DIA
    GEAR TOOTH ELEMENTS SHALL BE IN ACCORDANCE WITH EDI 9
REFERENCE
    ARC TOOTH THICKNESS AT PD = .2618 ±.0000
    BASE CIRCLE DIA = 3.7588

MATERIAL - AMS 6265 STEEL FORGED BARS

Figure 111.   Fatigue Test Gear Configuration 1—EX-78772.

149

BREAK TIP .005 MAX

.015 -.030 ROUND BREAK

16/ SURFACE E

.050 R MIN

40/ BEFORE SHOT PEENING

**ENLARGED VIEW OF GEAR PROFILE**
**SCALE NONE**

TOOTH SPACE SHALL BE FULLY GROUND INCLUDING ROOT
DIA AND ADJACENT FILLETS AFTER HEAT TREAT. NO
DISCONTINUITY SHALL OCCUR AT THE BLEND OF THE FILLET
RADIUS WITH THE ROOT DIA AND INVOLUTE SURFACES. SHOT
PEEN THE ENTIRE TOOTH SPACE PER EPS 12140 FOLLOWED BY
EPS 12196 AFTER GRINDING, THEN HONE SURFACE E TO VALUE SHOWN.
REMAINING SURFACES MAY BE PEENED PER EPS 12140 UNLESS
SPECIFICALLY CONTROLED BY A √ SYMBOL. STOCK REMOVAL BY
GRINDING TO BE UNIFORM ON PROFILE, FILLET RADIUS AND ROOT
DIA. WITHIN .002. MEASURE AND RECORD FILLET RADIUS, DISTANCE
OVER PINS AND ROOT DIA. BEFORE AND AFTER GRINDING.

REMOVE 12 TEETH SPACED AS SHOWN AFTER
GEAR TEETH ARE TO FINISHED SIZE AND
SIDE B    ALL GEAR TOOTH INSPECTION REQUIREMENTS
ARE COMPLETED

DIA A SHALL BE CONCENTRIC WITH PD WITHIN .002 TIR

BREAK SHARP EDGES .010 UOS

MACHINE ALL OVER.

SURFACE CHARACTERISTICS NOT CONTROLLED BY A √
SYMBOL SHALL BE COMMENSURATE WITH GOOD MANU-
FACTURING PRACTICES WHICH PRODUCE ACCEPTABLE
QUALITY LEVELS.

HEAT TREAT PER EPS 202

EMICAL ETCH POSITION NUMBERS
S SHOWN. RECORD INVOLUTE PROFILE
CHECKS FOR SIDE A OF TEETH
AND FOR SIDE B OF TEETH XI,
X4. RECORD POSITION NUMBERS
O TOOTH SPACING ERROR

CASE HARDEN GEAR TEETH OUTSIDE 3.340 DIA
(OPTIONAL TO CASE HARDEN ALL OVER) EFFECTIVE CASE
DEPTHS AS FOLLOWS:
     .035 -.045 BEFORE FINISHING
     .030 -.045 AFTER FINISHING
ROCKWELL HARDNESS — CASE C58 MIN
                     CORE C34 MIN

INSPECT PER EIS 985 (MAGNETIC)

NITAL ETCH PER EIS 1510 THEN
BLACK OXIDE PER AMS 2485

MATERIAL- AMS 6265 STEEL FORGED BARS

ALL DIMENSIONS TO BE MET AFTER PROCESSING
FORGING SHALL CONFORM TO EDI 138 AND EIS 502

+.0000
-.0046

WITH EDI 9

—EX-78772.

B

BREAK TIP .005 MAX

.015 -.030 ROUNDED BREAK

.025 R MIN

SURFACE E

BEFORE
SHOT PEENING

ENLARGED VIEW OF GEAR PROFILE
SCALE NONE

TOOTH SPACE SHALL BE FULLY GROUND INCLUDING ROOT
DIA AND ADJACENT FILLETS AFTER HEAT TREAT. NO
DISCONTINUITY SHALL OCCUR AT THE BLEND OF THE FILLET
RADIUS WITH THE ROOT DIA AND INVOLUTE SURFACES. SH
PEEN THE ENTIRE TOOTH SPACE PER EPS 12140 FOLLOWE
BY EPS 12176 AFTER GRINDING THEN HONE SURFACE E TO VA.
SHOWN. REMAINING SURFACES MAY BE PEENED PER EPS 12140 UN
SPECIFICALLY CONTROLED BY A √ SYMBOL. STOCK REMOVAL
GRINDING TO BE UNIFORM ON PROFILE, FILLET RADIUS AND RO
DIA. WITHIN .002. MEASURE AND RECORD FILLET RADIUS,
DISTANCE OVER PINS AND ROOT DIA. BEFORE AND AFTER GRIND

.252
.248

32

.7505 DIA A
.7500

.958 TYPICAL
.948 12 PLACES

SIDE A

SIDE B

B1

CSK 90° TO .810 DIA
(BOTH SIDES)

ELECTRO CHEMICAL ETCH ALLISON PART NO.
AND LAST CHANGE LETTER, "SER" AND SERIAL
NO. HERE PER AS 478-7A1 OR 7A2

REMOVE 12 TEETH SPACED AS SHOWN
AFTER GEAR TEETH ARE TO FINISHED
SIZE AND ALL GEAR TOOTH INSPECTION
B2  REQUIREMENTS ARE COMPLETED

ELECTRO CHEMICAL ETCH POSITION NUMBERS
ON TEETH AS SHOWN. RECORD INVOLUTE
PROFILE AND LEAD CHECKS FOR SIDE A
OF TEETH 1,2,3, AND 4; AND FOR SIDE B
OF TEETH X1, X2, X3 AND X4. RECORD POSITION
NUMBERS ON TOOTH TO TOOTH SPACING
ERROR CHECKS

MATERIAL - AMS 6265 STEEL FORGED BARS

| GEAR TOOTH CONTROL | | | |
|---|---|---|---|
| INVOLUTE PROFILE TOLERANCE | | | |
| SIDE A | OD | SIDE B | |
| 0.9° .0000 | ODB | | |
| 1.0 U -.0002 | | | |
| 12.0° | PD | SAME | |
| 13.4 U | | | |
| 23.1° .0000 | AFCA | | |
| 25.8 U -.0002 | | | |
| 25.0° | APD | | |
| 27.9 U | | | |
| 32.9° | BCD | | |
| 36.7 U | | | |
| SPACING TOLERANCE | | | |
| A .0002 | B .0002 | | |
| LEAD TOLERANCE | | | |
| A ±.0001 | B ±.0001 | | |
| FULLNESS TOLERANCE | | | |
| A .0000 | B .0000 | | |
| MAXIMUM HOLLOW IN FORM .0001 | | | |
| NOTE | 1 UNIT = .0147 IN | | |

SPUR GEAR DATA
  12 PITCH 24 TEETH
  20° PRESSURE ANGLE
  DISTANCE OVER TWO .1440 DIA PINS = 2.2000 +.0000 -.0047
    ROOT DIA = 1.792 ±.002
    PITCH DIA = 2.0000
    OUTSIDE DIA = 2.1667 +.0000
  ACTIVE PROFILE OUTSIDE 1.8969 DIA
GEAR TOOTH ELEMENTS SHALL BE IN ACCORDANCE WITH EDI 9
REFERENCE
  ARC TOOTH THICKNESS AT PD = .1309 +.0000
  BASE CIRCLE DIA = 1.8794

Figure 112.  Fatigue Test Gear Configuration 2—EX-78773.

151

A

BREAK TIP .005 MAX

-.030 ROUNDED BREAK

.025 R MIN

$^{16}/$ SURFACE E

$^{40}/$ BEFORE SHOT PEENING

ENLARGED VIEW OF GEAR PROFILE
SCALE NONE

OTH SPACE SHALL BE FULLY GROUND INCLUDING ROOT
A AND ADJACENT FILLETS AFTER HEAT TREAT. NO
S CONTINUITY SHALL OCCUR AT THE BLEND OF THE FILLET
DIUS WITH THE ROOT DIA AND INVOLUTE SURFACES. SHOT
EN THE ENTIRE TOOTH SPACE PER EPS 12140 FOLLOWED
EPS 12176 AFTER GRINDING THEN HONE SURFACE E TO VALUE
OWN. REMAINING SURFACES MAY BE PEENED PER EPS 12140 UNLESS
ECIFICALLY CONTROLED BY A √ SYMBOL. STOCK REMOVAL BY
INDING TO BE UNIFORM ON PROFILE, FILLET RADIUS AND ROOT
9. WITHIN .002. MEASURE AND RECORD FILLET RADIUS,
TANCE OVER PINS AND ROOT DIA. BEFORE AND AFTER GRINDING.

.958 TYPICAL
.948 12 PLACES

SIDE A

SIDE B

REMOVE 12 TEETH SPACED AS SHOWN
AFTER GEAR TEETH ARE TO FINISHED
SIZE AND ALL GEAR TOOTH INSPECTION
REQUIREMENTS ARE COMPLETED

ELECTRO CHEMICAL ETCH POSITION NUMBERS
ON TEETH AS SHOWN. RECORD INVOLUTE
PROFILE AND LEAD CHECKS FOR SIDE A
OF TEETH 1,2,3 AND 4; AND FOR SIDE B
OF TEETH X1, X2, X3 AND X4. RECORD POSITION
NUMBERS ON TOOTH TO TOOTH SPACING
ERROR CHECKS

MATERIAL - AMS 6265 STEEL FORGED BARS

DIA A SHALL BE CONCENTRIC WITH PD WITHIN .002 TIR

BREAK SHARP EDGES .010 UOS

MACHINE ALL OVER.

SURFACE CHARACTERISTICS NOT CONTROLLED BY A √
SYMBOL SHALL BE COMMENSURATE WITH GOOD MANU-
FACTURING PRACTICES WHICH PRODUCE ACCEPTABLE
QUALITY LEVELS.

HEAT TREAT PER EPS 202

CASE HARDEN GEAR TEETH OUTSIDE 1.570 DIA
(OPTIONAL TO CASE HARDEN ALL OVER) EFFECTIVE CASE
DEPTHS AS FOLLOWS:
      .020-.030 BEFORE FINISHING
      .015-.030 AFTER FINISHING
ROCKWELL HARDNESS - CASE C58 MIN
                    CORE C34 MIN

INSPECT PER EIS 985 (MAGNETIC)

NITAL ETCH PER EIS 1510 THEN
BLACK OXIDE PER AMS 2485

ALL DIMENSIONS TO BE MET AFTER PROCESSING
FORGING SHALL CONFORM TO EDI 138 AND EIS 502

$2000 ^{+.0000}_{-.0047}$

ANCE WITH EDI 9

8

2—EX-78773.

BREAK TIP .005 MAX

.015 -.030 ROUND BREAK

.080 R MIN

SURFACE

16/

40/ BEFO
SHOT PEE

ENLARGED VIEW OF GEAR PROFILE
SCALE NONE

TOOTH SPACE SHALL BE FULLY GROUND INCLUDING
DIA AND ADJACENT FILLETS AFTER HEAT TRE
DISCONTINUITY SHALL OCCUR AT THE BLEND OF T
RADIUS WITH THE ROOT DIA AND INVOLUTE SURI
PEEN THE ENTIRE TOOTH SPACE PER EPS 12140
EPS 12170 AFTER GRINDING, THEN HONE SURFACE E T
REMAINING SURFACES MAY BE PEENED PER EPS 12140
SPECIFICALLY CONTROLLED BY A √ SYMBOL. STOCK F
GRINDING TO BE UNIFORM ON PROFILE, FILLET RADIUS
WITHIN .002. MEASURE AND RECORD FILLET RADIUS
OVER PINS AND ROOT DIA. BEFORE AND AFTER GRINL

REMOVE 12 TEETH SPACED AS SHOWN AFTE
GEAR TEETH ARE TO FINISHED SIZE AND ALL
GEAR TOOTH INSPECTION REQUIREMENTS ARE
COMPLETED

.502
.498

1.910
1.900

SIDE A

1.5005
1.5000 DIA A

CSK 90° TO 1.560 DIA
(BOTH SIDES)

TYPICAL
12 PLACES

SIDE B

ELECTRO CHEMICAL ETCH ALLISON PART NO. AND
LAST CHANGE LETTER "SER" AND SERIAL NO. HERE
PER AS 478-7A1 OR 7A2

ELECTRO CHEMICAL ETCH POSITION NUMBERS ON TEETH
AS SHOWN. RECORD INVOLUTE PROFILE AND LEAD
CHECKS FOR SIDE A OF TEETH 1,2,3, AND 4; AND
FOR SIDE B OF TEETH X1, X2, X3 AND X4. RECORD
POSITION NUMBERS ON TOOTH TO TOOTH SPACING
ERROR CHECKS

DIA A S

BREAK

MACHINE

SURFACE CA
SYMBOL SH
FACTURING
QUALITY LE

HEAT TR

CASE H
(OPTION
DEPTHS
.0
.0
ROCKWE

INSPEC

NITAL E
BLACK

ALL DIM
FORGIN

MATERIAL- AMS 6265 STEEL FORGED BAR

| GEAR TOOTH CONTROL | | | |
|---|---|---|---|
| INVOLUTE PROFILE TOLERANCE | | | |
| SIDE A | | OD | SIDE B |
| 0.5° | 1.0000 | OD | |
| 1.0 U | 1.0002 | ODB | |
| 12.0° | | PD | SAME |
| 26.8 U | | | |
| 23.6° | .0000 | MFCA | |
| 32.6 U | 1.0002 | | |
| 24.5° | | APD | |
| 34.7 U | | | |
| 32.9° | | BCD | |
| 73.3 U | | | |
| SPACING TOLERANCE | | | |
| A .0002 | | B .0002 | |
| LEAD TOLERANCE | | | |
| A ±.0001 | | B ±.0001 | |
| FULLNESS TOLERANCE | | | |
| A .0000 | | B .0000 | |
| MAXIMUM HOLLOW IN FORM .0001 | | | |
| NOTE | 1 UNIT = .0147 IN. | | |

SPUR GEAR DATA
    6 PITCH 24 TEETH
    20° PRESSURE ANGLE
    DISTANCE OVER TWO .2880 DIA PINS- 4.3999 +.0000 -.0046
        ROOT DIA- 3.583 ±.002
        PITCH DIA- 4.0000
        OUTSIDE DIA- 4.3333 +.0000 -.0000
    ACTIVE PROFILE OUTSIDE 3.7984 DIA
    GEAR TOOTH ELEMENTS SHALL BE IN ACCORDANCE WITH EDI 9
REFERENCE
    ARC TOOTH THICKNESS AT PD- .2618 +.0000 -.0000
    BASE CIRCLE DIA- 3.7588

Figure 113.   Fatigue Test Gear Configuration 3—EX-78774.

153

A

BREAK TIP .005 MAX

.015 -.030 ROUND BREAK

16/ — SURFACE E

.080 R MIN

40/ — BEFORE
SHOT PEENING

ENLARGED VIEW OF GEAR PROFILE
SCALE NONE

TOOTH SPACE SHALL BE FULLY GROUND INCLUDING ROOT
DIA AND ADJACENT FILLETS AFTER HEAT TREAT. NO
DISCONTINUITY SHALL OCCUR AT THE BLEND OF THE FILLET
RADIUS WITH THE ROOT DIA AND INVOLUTE SURFACES. SHOT
PEEN THE ENTIRE TOOTH SPACE PER EPS 12140 FOLLOWED BY
EPS 12170 AFTER GRINDING, THEN HONE SURFACE E TO VALUE SHOWN.
REMAINING SURFACES MAY BE PEENED PER EPS 12140 UNLESS
SPECIFICALLY CONTROLLED BY A √ SYMBOL. STOCK REMOVAL BY
GRINDING TO BE UNIFORM ON PROFILE, FILLET RADIUS AND ROOT DIA.
WITHIN .002. MEASURE AND RECORD FILLET RADIUS, DISTANCE
OVER PINS AND ROOT DIA. BEFORE AND AFTER GRINDING.

REMOVE 12 TEETH SPACED AS SHOWN AFTER
GEAR TEETH ARE TO FINISHED SIZE AND ALL
GEAR TOOTH INSPECTION REQUIREMENTS ARE
COMPLETED

SIDE B

DIA A SHALL BE CONCENTRIC WITH PD WITHIN .002 TIR

BREAK SHARP EDGES .010 UOS

MACHINE ALL OVER.

ELECTRO CHEMICAL ETCH POSITION NUMBERS ON TEETH
AS SHOWN. RECORD INVOLUTE PROFILE AND LEAD
CHECKS FOR SIDE A OF TEETH 1,2,3, AND 4; AND
FOR SIDE B OF TEETH X1, X2, X3 AND X4. RECORD
POSITION NUMBERS ON TOOTH TO TOOTH SPACING
ERROR CHECKS

SURFACE CHARACTERISTICS NOT CONTROLLED BY A √
SYMBOL SHALL BE COMMENSURATE WITH GOOD MANU-
FACTURING PRACTICES WHICH PRODUCE ACCEPTABLE
QUALITY LEVELS.

HEAT TREAT PER EPS 202

CASE HARDEN GEAR TEETH OUTSIDE 3.340 DIA
(OPTIONAL TO CASE HARDEN ALL OVER) EFFECTIVE CASE
DEPTHS AS FOLLOWS:
      .035 -.045 BEFORE FINISHING
      .030 -.045 AFTER FINISHING
ROCKWELL HARDNESS – CASE C58 MIN
                   CORE C34 MIN

INSPECT PER EIS 985 (MAGNETIC)

MATERIAL- AMS 6265 STEEL FORGED BAR

NITAL ETCH PER EIS 1510 THEN
BLACK OXIDE PER AMS 2485

ALL DIMENSIONS TO BE MET AFTER PROCESSING
FORGING SHALL CONFORM TO EDI 138 AND EIS 502

3999 +.0000 -.0046

4
?DANCE WITH EDI 9
388

3—EX-78774.

B

BREAK TIP .005 MAX

.015 -.030 ROUNDED BREAK

SURFACE E

.040 R MIN

BEFORE
SHOT PEENING

**ENLARGED VIEW OF GEAR PROFILE
SCALE NONE**

TOOTH SPACE SHALL BE FULLY GROUND INCLUDING ROOT
DIA AND ADJACENT FILLETS AFTER HEAT TREAT. NO
DISCONTINUITY SHALL OCCUR AT THE BLEND OF THE FILLET
RADIUS WITH THE ROOT DIA AND INVOLUTE SURFACES. SHOT
PEEN THE ENTIRE TOOTH SPACE PER EPS 12140 FOLLOWED BY
EPS 12176 AFTER GRINDING, THEN HONE SURFACE E TO VALUE
SHOWN. REMAINING SURFACES MAY BE PEENED PER EPS 12140
UNLESS SPECIFICALLY CONTROLLED BY A √ SYMBOL. STOCK
REMOVAL BY GRINDING TO BE UNIFORM ON PROFILE, FILLET
RADIUS AND ROOT DIA WITHIN .002. MEASURE AND RECORD
FILLET RADIUS, DISTANCE OVER PINS, AND ROOT DIA BEFORE
AND AFTER GRINDING.

.252
.248

.7505 DIA A
.7500

.958 TYPICAL
.948 12 PLACES

SIDE B

CSK 90° TO .810 DIA
(BOTH SIDES)

REMOVE 12 TEETH SPACED AS
SHOWN AFTER GEAR TEETH ARE TO
FINISHED SIZE AND ALL GEAR TOOTH
INSPECTION REQUIREMENTS ARE
COMPLETED

SIDE A

DIA A SHA

BREAK SH

MACHINE

ELECTRO CHEMICAL ETCH ALLISON PART NO.
AND LAST CHANGE LETTER, "SER" AND SERIAL NO.
HERE PER AS 478-7A1 OR 72

ELECTRO CHEMICAL ETCH POSITION NUMBERS
ON TEETH AS SHOWN. RECORD INVOLUTE
PROFILE AND LEAD CHECKS FOR SIDE A
OF TEETH 1,2,3, AND 4; AND FOR SIDE B
OF TEETH X1, X2, X3, AND X4. RECORD POSITION
NUMBERS ON TOOTH TO TOOTH SPACING
ERROR CHECKS

SURFACE CHAR
SYMBOL SHALL
FACTURING PRA
QUALITY LEVEL

HEAT TRE

CASE HAF
(OPTIONAL
DEPTHS
.020
.015
ROCKWELL

MATERIAL - AMS 6265 STEEL FORGED BARS

INSPECT

NITAL ETC
BLACK OX

ALL DIMENS
FORGINGS S

| GEAR TOOTH CONTROL | | | | |
|---|---|---|---|---|
| **INVOLUTE PROFILE TOLERANCE** | | | | |
| SIDE A | | OD | SIDE B | |
| .9° | .0000 | ODB | | |
| 1.0 U | -.0002 | | | |
| 12.0° | | PD | SAME | |
| 13.4 U | | | | |
| 23.1° | .0000 | HFCA | | |
| 25.8 U | -.0002 | | | |
| 25.0° | | APD | | |
| 27.8 U | | | | |
| 32.9° | | BCD | | |
| 34.7 U | | | | |

| SPACING TOLERANCE | |
|---|---|
| A .0002 | B .0002 |

| LEAD TOLERANCE | |
|---|---|
| A ±.0001 | B ±.0001 |

| FULLNESS TOLERANCE | |
|---|---|
| A .0000 | B .0000 |

MAXIMUM HOLLOW IN FORM .0001

| NOTE | 1 UNT• .0047 IN |
|---|---|

SPUR GEAR DATA
12 PITCH 24 TEETH
20° PRESSURE ANGLE
DISTANCE OVER TWO .1440 DIA PINS• 2.2000 +.0000 -.0047
ROOT DIA• 1.792 ±.062
PITCH DIA• 2.0000
OUTSIDE DIA• 2.1667 +.0000
ACTIVE PROFILE OUTSIDE 1.8969 DIA
GEAR TOOTH ELEMENTS SHALL BE IN ACCORDANCE WITH EDI 9
REFERENCE
ARC TOOTH THICKNESS AT PD• .1309 +.0000
BASE CIRCLE DIA• 1.8794

**Figure 114. Fatigue Test Gear Configuration 4—EX-78775.**

155

A

BREAK TIP .005 MAX

-.030 ROUNDED BREAK

SURFACE E

.040 R MIN

BEFORE
SHOT PEENING

**ENLARGED VIEW OF GEAR PROFILE**
**SCALE NONE**

OTH SPACE SHALL BE FULLY GROUND INCLUDING ROOT
A AND ADJACENT FILLETS AFTER HEAT TREAT. NO
SCONTINUITY SHALL OCCUR AT THE BLEND OF THE FILLET
DIUS WITH THE ROOT DIA AND INVOLUTE SURFACES. SHOT
EN THE ENTIRE TOOTH SPACE PER EPS 12140 FOLLOWED BY
S 12176 AFTER GRINGING, THEN HONE SURFACE E TO VALUE
OWN. REMAINING SURFACES MAY BE PEENED PER EPS 12140
LESS SPECIFICALLY CONTROLLED BY A √ SYMBOL. STOCK
EMOVAL BY GRINDING TO BE UNIFORM ON PROFILE, FILLET
DIUS AND ROOT DIA WITHIN .002. MEASURE AND RECORD
LET RADIUS, DISTANCE OVER PINS, AND ROOT DIA BEFORE
O AFTER GRINDING.

.958 TYPICAL
.948 12 PLACES

12 TEETH SPACED AS
AFTER GEAR TEETH ARE TO
ED SIZE AND ALL GEAR TOOTH
TION REQUIREMENTS ARE
ETED

SIDE B

SIDE A

RO CHEMICAL ETCH POSITION NUMBERS
ETH AS SHOWN RECORD INVOLUTE
LE AND LEAD CHECKS FOR SIDE A
ETH 1,2,3, AND 4; AND FOR SIDE B
ETH X1, X2, X3, AND X4. RECORD POSITION
ERS ON TOOTH TO TOOTH SPACING
OR CHECKS

DIA A SHALL BE CONCENTRIC WITH PD WITHIN .002 TIR

BREAK SHARP EDGES .010 UOS

MACHINE ALL OVER.

SURFACE CHARACTERISTICS NOT CONTROLLED BY A √
SYMBOL SHALL BE COMMENSURATE WITH GOOD MANU-
FACTURING PRACTICES WHICH PRODUCE ACCEPTABLE
QUALITY LEVELS.

HEAT TREAT PER EPS 202

CASE HARDEN GEAR TEETH OUTSIDE 1.570 DIA
(OPTIONAL TO CASE HARDEN ALL OVER) EFFECTIVE CASE
DEPTHS AS FOLLOWS:
    .020-.030 BEFORE FINISHING
    .015-.030 AFTER FINISHING
ROCKWELL HARDNESS - CASE C58 MIN
                    CORE C34 MIN

INSPECT PER EIS 985 (MAGNETIC)

NITAL ETCH PER EIS 1510 THEN
BLACK OXIDE PER AMS 2485

ALL DIMENSIONS TO BE MET AFTER PROCESSING
FORGINGS SHALL CONFORM TO EDI 138 AND EIS 502

MATERIAL-AMS 6265 STEEL FORGED BARS

.2000 +.0000 -.0047

A
RDANCE WITH EDI 9

388
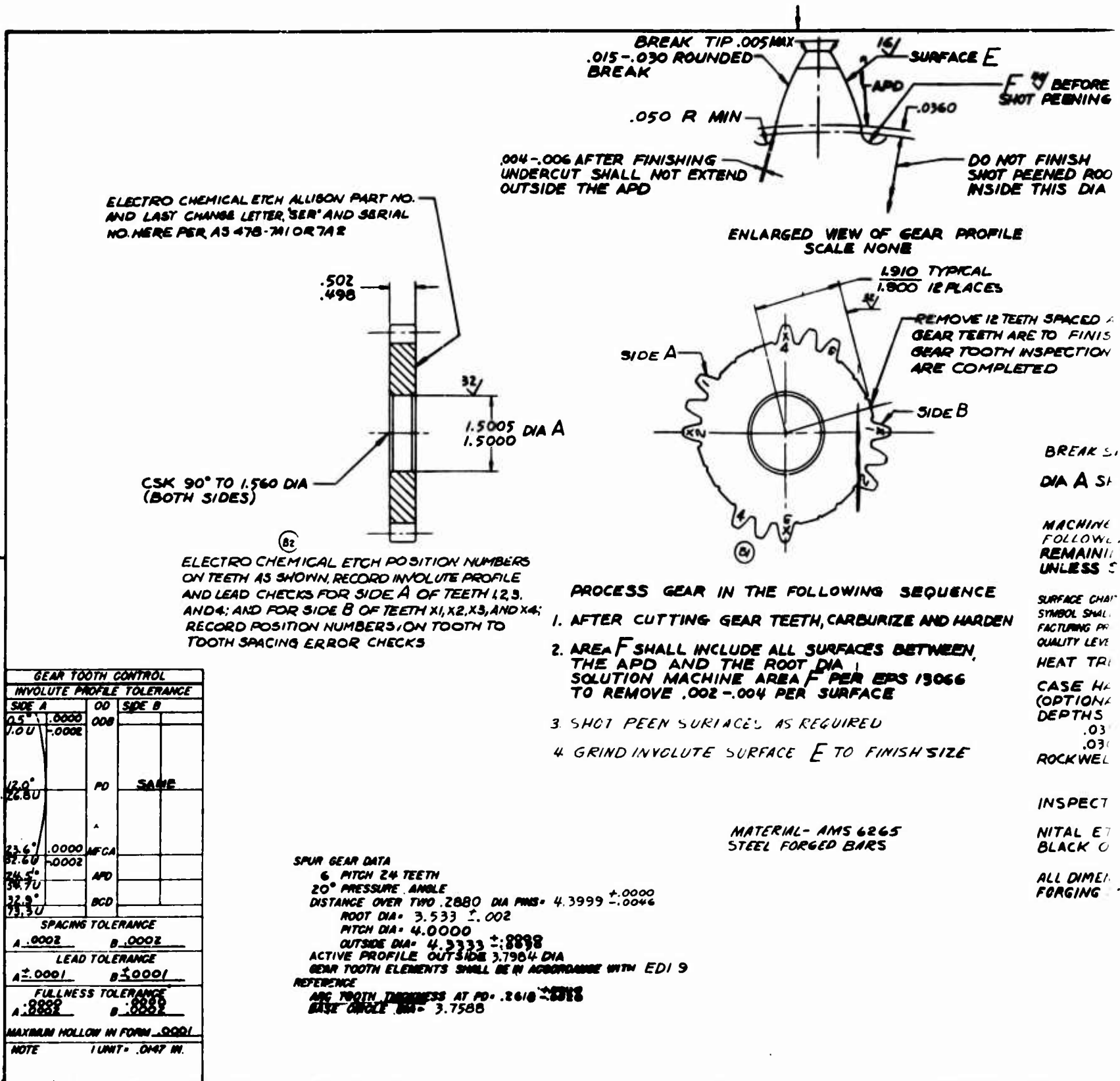
Figure 115. Fatigue Test Gear Configuration 5—EX-78776.

157

A

BREAK TIP .005 MAX
.015-.030 ROUNDED
BREAK

SURFACE E

APD

F ⍍ BEFORE
SHOT PEENING

.050 R MIN

.0360

-.006 AFTER FINISHING
ERCUT SHALL NOT EXTEND
SIDE THE APD

DO NOT FINISH
SHOT PEENED ROOT
INSIDE THIS DIA

**ENLARGED VIEW OF GEAR PROFILE**
SCALE NONE

1.910 TYPICAL
1.900 12 PLACES

REMOVE 12 TEETH SPACED AS SHOWN AFTER
GEAR TEETH ARE TO FINISHED SIZE AND ALL
GEAR TOOTH INSPECTION REQUIREMENTS
ARE COMPLETED

SIDE A

SIDE B

BREAK SHARP EDGES .010 UOS

DIA A SHALL BE CONCENTRIC WITH PD WITHIN .002 TIR

MACHINE ALL OVER. PEEN GEAR TEETH PER EPS 12140
FOLLOWED BY EPS 12176
REMAINING SURFACES MAY BE PEENED PER EPS 12140
UNLESS SPECIFICALLY CONTROLLED BY A √ SYMBOL

PROCESS GEAR IN THE FOLLOWING SEQUENCE

1. AFTER CUTTING GEAR TEETH, CARBURIZE AND HARDEN

2. AREA F SHALL INCLUDE ALL SURFACES BETWEEN
   THE APD AND THE ROOT DIA.
   SOLUTION MACHINE AREA F PER EPS 13066
   TO REMOVE .002-.004 PER SURFACE

3. SHOT PEEN SURFACES AS REQUIRED

4. GRIND INVOLUTE SURFACE E TO FINISH SIZE

MATERIAL- AMS 6265
STEEL FORGED BARS

SURFACE CHARACTERISTICS NOT CONTROLLED BY A √
SYMBOL SHALL BE COMMENSURATE WITH GOOD MANU-
FACTURING PRACTICES WHICH PRODUCE ACCEPTABLE
QUALITY LEVELS.

HEAT TREAT PER EPS 202

CASE HARDEN GEAR TEETH OUTSIDE 3.340 DIA
(OPTIONAL TO CASE HARDEN ALL OVER) EFFECTIVE CASE
DEPTHS AS FOLLOWS:
    .035-.045 BEFORE FINISHING
    .030-.045 AFTER FINISHING
ROCKWELL HARDNESS - CASE C58 MIN
                      CORE C34 MIN

INSPECT PER EIS 985 (MAGNETIC)

NITAL ETCH PER EIS 1510 THEN
BLACK OXIDE PER AMS 2485

ALL DIMENSIONS TO BE MET AFTER PROCESSING
FORGING SHALL CONFORM TO EDI 138 AND EIS 502

$4.3999 ^{+.0000}_{-.0046}$

38
4 DIA
ACCORDANCE WITH EDI 9

B
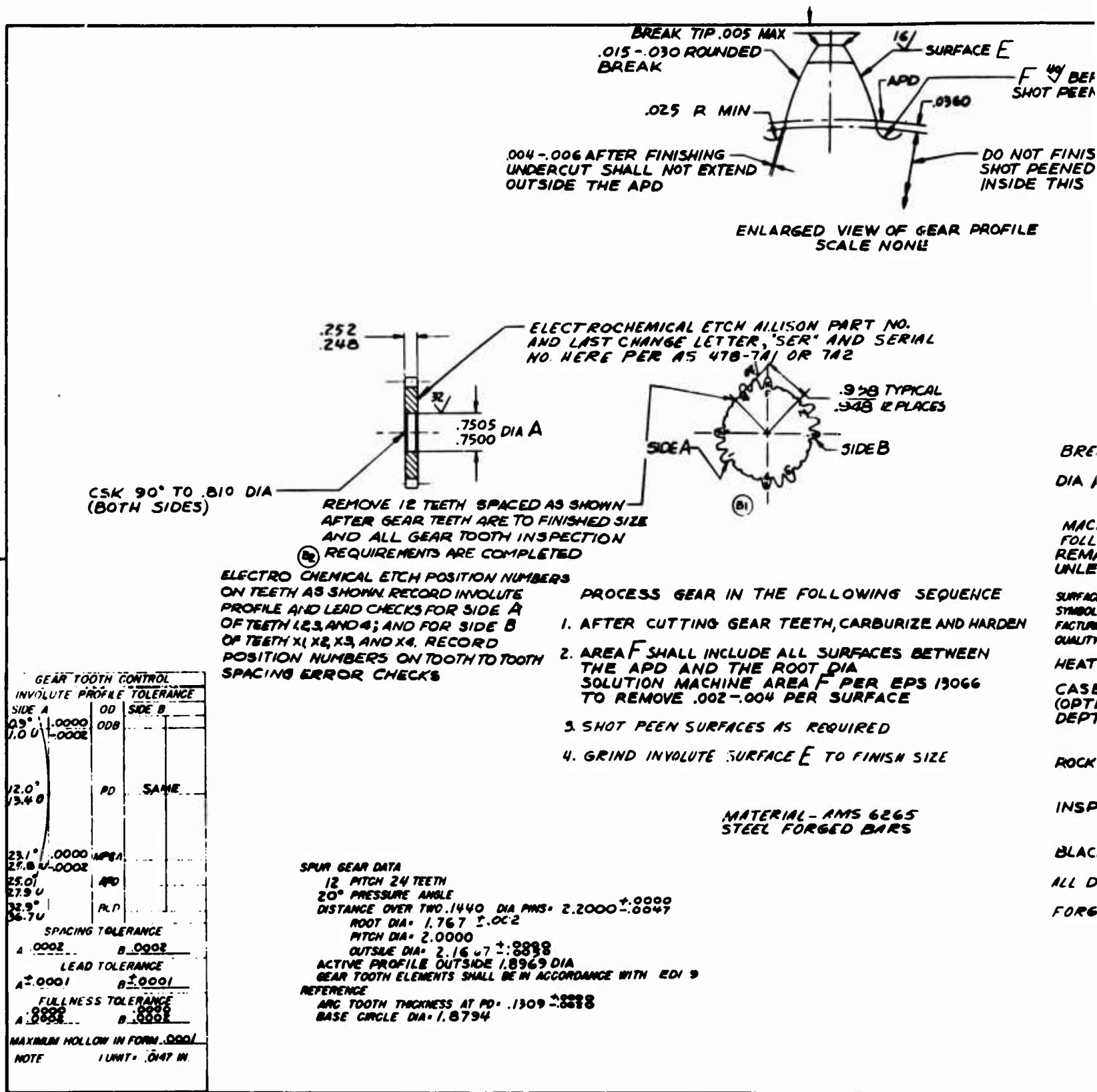
Figure 116.   Fatigue Test Gear Configuration 6—EX-78777.

159

A

IP .005 MAX
-OUNDED-
-SURFACE E

16/

F BEFORE
SHOT PEENING

APD

.0360

₂ MIN

DO NOT FINISH
SHOT PEENED ROOT
INSIDE THIS DIA

NG
XTEND
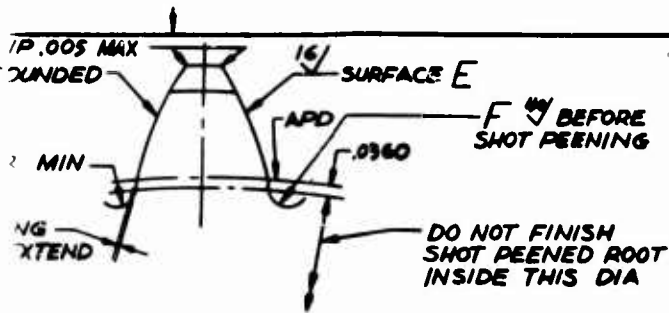
**ENLARGED VIEW OF GEAR PROFILE**
**SCALE NONE**

TCH ALLISON PART NO.
ETTER, "SER" AND SERIAL
478-7A/ OR 7A2

.958 TYPICAL
.948 12 PLACES

SIDE B

(B1)

LESS ⸱ ᐧᐧR IN THE FOLLOWING SEQUENCE

FACE Cʰ⸱ ᐧ⸱ᐧ ᐧIG GEAR TEETH, CARBURIZE AND HARDEN

INCLUDE ALL SURFACES BETWEEN
D THE ROOT DIA
AT TH⸱ ⸱ᐧ CHINE AREA F PER EPS 13066
002 -.004 PER SURFACE
SE H
TION⸱⸱ ⸱ IRFACES AS REQUIRED
PTHS
⸱⸱⸱ E SURFACE E TO FINISH SIZE

MATERIAL – AMS 6265
STEEL FORGED BARS

BREAK SHARP EDGES .010 UOS

DIA A SHALL BE CONCENTRIC WITH PD WITHIN .002 TIR

MACHINE ALL OVER. PEEN GEAR TEETH PER EPS 12140
FOLLOWED BY EPS 12176
REMAINING SURFACES MAY BE PEENED PER EPS 12140
UNLESS SPECIFICALLY CONTROLLED BY A √ SYMBOL

SURFACE CHARACTERISTICS NOT CONTROLLED BY A √
SYMBOL SHALL BE COMMENSURATE WITH GOOD MANU-
FACTURING PRACTICES WHICH PRODUCE ACCEPTABLE
QUALITY LEVELS.

HEAT TREAT PER EPS 202

CASE HARDEN GEAR TEETH OUTSIDE 1.570 DIA
(OPTIONAL TO CASE HARDEN ALL OVER) EFFECTIVE CASE
DEPTHS AS FOLLOWS:
   .020-.030  BEFORE FINISHING
   .015-.030  AFTER FINISHING
ROCKWELL HARDNESS – CASE C58 MIN
                            CORE C 34 MIN

INSPECT PER EIS 985 (MAGNETIC)

BLACK OXIDE PER AMS 2485

ALL DIMENSIONS TO BE MET AFTER PROCESSING

FORGINGS SHALL CONFORM TO EDI 138 AND EIS 502

777.

B

BREAK TIP .005 MAX
.015 -.030 ROUNDED BREAK
.080 R MIN
.004 -.006 AFTER FINISHING
UNDERCUT SHALL NOT EXTEND
OUTSIDE THE APD

16/ SURFACE E
APD
.0360
F 9/ BEFORE SHOT PEENING
DO NOT FINISH SHOT PEENED ROOT INSIDE THIS DIA

ENLARGED VIEW OF GEAR PROFILE
SCALE NONE

1.910 TYPICAL
1.910 12 PLACES

ELECTRO CHEMICAL ETCH ALLISON PART NO.
AND LAST CHANGE LETTER "SER" AND SERIAL
NO. HERE PER AD 478-741 OR 742

.502
.498

32/

1.5005 DIA A
1.5000

CSK 90° TO 1.560 DIA
(BOTH SIDES)

(B2)

ELECTRO CHEMICAL ETCH POSITION NUMBERS ON TEETH
AS SHOWN. RECORD INVOLUTE PROFILE AND LEAD
CHECKS FOR SIDE A OF TEETH 1,2,3, AND 4; AND FOR
SIDE B OF TEETH X1, X2, X3, AND X4. RECORD POSITION
NUMBERS ON TOOTH TO TOOTH SPACING ERROR
CHECKS

SIDE A
SIDE B

REMOVE 12 TEETH SPACED
GEAR TEETH ARE TO FINISH
GEAR TOOTH INSPECTION RE
COMPLETED

(B1)

PROCESS GEAR IN THE FOLLOWING SEQUENCE

1. AFTER CUTTING GEAR TEETH, CARBURIZE AND HARDEN

2. AREA F SHALL INCLUDE ALL SURFACES BETWEEN
THE APD AND THE ROOT DIA
SOLUTION MACHINE AREA F PER EPS 13066
TO REMOVE .002 -.004 PER SURFACE

3. SHOT PEEN SURFACES AS REQUIRED

4. GRIND INVOLUTE SURFACE F TO FINISH SIZE

MATERIAL- AMS 6265
STEEL FORGED BARS

BREAK S
DIA A SHA
MACHINE
FOLLOWED
REMAININ
UNLESS SF
SURFACE CHAR
SYMBOL SHALL
FACTURING PRA
QUALITY LEVEL
HEAT TREAT
CASE HAR
(OPTIONAL
DEPTHS
.035
.030
ROCKWELL
INSPECT
BLACK OX
ALL DIMEN
FORGINGS

## GEAR TOOTH CONTROL

### INVOLUTE PROFILE TOLERANCE

| SIDE A | | OD | SIDE B |
|--------|--------|------|--------|
| 10.5° 7.0 U | .0000 -.0002 | OOB | |
| 12.0° 26.8 U | | PD | SAME |
| 23.6° 32.6 U | .0000 -.0002 | AFCA | |
| 34.5° 34.9 U | | APD | |
| 32.9° 39.3 U | | BCD | |

### SPACING TOLERANCE
A .0003   B .0003

### LEAD TOLERANCE
A ±.0001   B ±.0001

### FULLNESS TOLERANCE
A .0000   .0000

MAXIMUM HOLLOW IN FORM .0001

NOTE   1 UNIT = .0147 IN

SPUR GEAR DATA
6 PITCH 24 TEETH
20° PRESSURE ANGLE
DISTANCE OVER TWO .2880 DIA PINS = 4.3999 ±.0000
ROOT DIA = 3.533 ±.002
PITCH DIA = 4.0000
OUTSIDE DIA = 4.3333 ±.0000
ACTIVE PROFILE OUTSIDE 3.7984 DIA
GEAR TOOTH ELEMENTS SHALL BE IN ACCORDANCE WITH EDI 9
REFERENCE
ARC TOOTH THICKNESS AT PD = .2618 ±.0000
BASE CIRCLE DIA = 3.7588

Figure 117.   Fatigue Test Gear Configuration 7—EX-78778.

161

A

MAX

16/ SURFACE E

APD

E ⁴/ BEFORE
SHOT PEENING

.0360

DO NOT FINISH
SHOT PEENED ROOT
INSIDE THIS DIA

LARGED VIEW OF GEAR PROFILE
SCALE NONE

1.910 TYPICAL
1.900 12 PLACES

36/

REMOVE 12 TEETH SPACED AS SHOWN AFTER
GEAR TEETH ARE TO FINISHED SIZE AND ALL
GEAR TOOTH INSPECTION REQUIREMENTS ARE
COMPLETED

SIDE B

X 4

BREAK SHARP EDGES .010 UOS

DIA A SHALL BE CONCENTRIC WITH PD WITHIN .002 TIR

MACHINE ALL OVER. PEEN GEAR TEETH PER EPS 12140
FOLLOWED BY EPS 12176
REMAINING SURFACES MAY BE PEENED PER EPS 12140
UNLESS SPECIFICALLY CONTROLLED BY A √ SYMBOL

HE FOLLOWING SEQUENCE

R TEETH, CARBURIZE AND HARDEN

E ALL SURFACES BETWEEN
ROOT DIA
AREA F PER EPS 13066
04 PER SURFACE

AS REQUIRED

CE E TO FINISH SIZE

ATERIAL- AMS 6265
EEL FORGED BARS

SURFACE CHARACTERISTICS NOT CONTROLLED BY A √
SYMBOL SHALL BE COMMENSURATE WITH GOOD MANU-
FACTURING PRACTICES WHICH PRODUCE ACCEPTABLE
QUALITY LEVELS.

HEAT TREAT PER EPS 202

CASE HARDEN GEAR TEETH OUTSIDE 3.340 DIA
(OPTIONAL TO CASE HARDEN ALL OVER) EFFECTIVE CASE
DEPTHS AS FOLLOWS:
.035 -.045 BEFORE FINISHING
.030 -.045 AFTER FINISHING
ROCKWELL HARDNESS - CASE C58 MIN
CORE C34 MIN

INSPECT PER EIS 985 (MAGNETIC)

BLACK OXIDE PER AMS 2485

ALL DIMENSIONS TO BE MET AFTER PROCESSING

FORGINGS SHALL CONFORM TO ED1138 AND EIS 502

B

ENLARGED VIEW OF GEAR PROFILE
SCALE NONE

BREAK TIP .005 MAX
.015 -.030 ROUNDED BREAK
SURFACE E
F ⱽ BEF SHOT PEEN
APD
.0360
.040 R MIN
.004 -.006 AFTER FINISHING
UNDERCUT SHALL NOT EXTEND
OUTSIDE THE APD
DO NOT FINIS SHOT PEENED INSIDE THIS

ELECTROCHEMICAL ETCH ALLISON PART NO. AND LAST CHANGE LETTER, "SER" AND SERIAL NO. HERE PER AS 478-7A1 OR 7A2

.252
.248

32 /

.7505 DIA A
.7500

CSK 90° TO .810 DIA
(BOTH SIDES)

ELECTRO CHEMICAL ETCH POSITION NUMBERS ON TEETH AS SHOWN, RECORD INVOLUTE PROFILE AND LEAD CHECKS FOR SIDE A OF TEETH 1, 2, 3, AND 4; AND FOR SIDE B OF TEETH X1, X2, X3, AND X4. RECORD POSITION NUMBERS ON TOOTH TO TOOTH SPACING ERROR CHECKS

.958 TYPICAL
.948 12 PLACES

SIDE B
SIDE A

REMOVE 12 TEETH SPACED AS SHOWN AFTER GEAR TEETH ARE TO FINISHED SIZE AND ALL GEAR TOOTH INSPECTION REQUIREMENTS ARE COMPLETED

PROCESS GEAR IN THE FOLLOWING SEQUENCE

1. AFTER CUTTING GEAR TEETH, CARBURIZE AND HARDEN

2. AREA F SHALL INCLUDE ALL SURFACES BETWEEN THE APD AND THE ROOT DIA SOLUTION MACHINE AREA F PER EPS 13066 TO REMOVE .002 -.004 PER SURFACE

3. SHOT PEEN SURFACES AS REQUIRED

4. GRIND INVOLUTE SURFACE E TO FINISH SIZE

MATERIAL- AMS 6265
STEEL FORGED BARS

| GEAR TOOTH CONTROL | | | |
|---|---|---|---|
| INVOLUTE PROFILE TOLERANCE | | | |
| SIDE A | OD | SIDE B | |
| 9.9° / 7.0 U | .0000 -.0002 | OOB | |
| 12.0° / 13.4 0 | | PD | SAME |
| 23.1° / 25.8 0 | .0000 -.0002 | MPCA | |
| 25.0° / 27.9 0 | | APD | |
| 32.9° / 36.7 0 | | BCD | |

SPACING TOLERANCE
A .0002     B .0002

LEAD TOLERANCE
A ±.0001     B ±.0001

FULLNESS TOLERANCE
A .0000 / .0002     B .0000 / .0002

MAXIMUM HOLLOW IN FORM .0001
NOTE     1 UNIT = .0147 IN

SPUR GEAR DATA
12 PITCH 24 TEETH
20° PRESSURE ANGLE
DISTANCE OVER TWO .1440 DIA PINS = 2.2000 +.0000 -.0047
ROOT DIA = 1.767 ±.002
PITCH DIA = 2.0000
OUTSIDE DIA = 2.1667 +.0000 -.0000
ACTIVE PROFILE OUTSIDE 1.8969 DIA
GEAR TOOTH ELEMENTS SHALL BE IN ACCORDANCE WITH EDI 9
REFERENCE
ARC TOOTH THICKNESS AT PD = .1309 +.0000 -.0000
BASE CIRCLE DIA = 1.8794

Figure 118. Fatigue Test Gear Configuration 8—EX-78779.

163

P.005 MAX
ROUNDED ────── 16/ ── SURFACE E

APD ── F ⌄⁴ᵞ BEFORE
SHOT PEENING

R MIN ── .0360

HING ── DO NOT FINISH
EXTEND ── SHOT PEENED ROOT
INSIDE THIS DIA
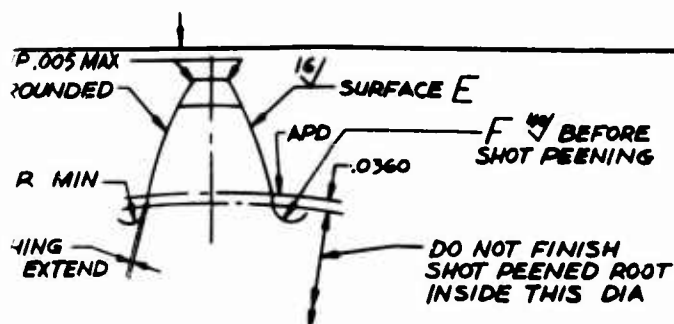
**ENLARGED VIEW OF GEAR PROFILE**
**SCALE NONE**

ETCH ALLISON PART NO. AND
TER, "SER" AND SERIAL NO. HERE
R 7A2
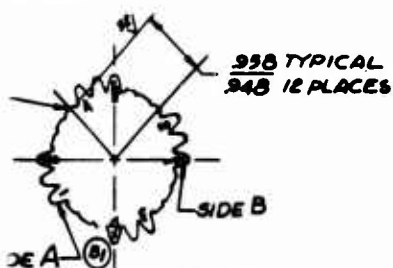
.998 TYPICAL
.948 12 PLACES

SIDE B

E A (8/)

EETH SPACED AS SHOWN
R TEETH ARE TO FINISHED
L GEAR TOOTH INSPECTION
NTS ARE COMPLETED

AR IN THE FOLLOWING SEQUENCE ··

ING GEAR TEETH, CARBURIZE AND HARDEN

. INCLUDE ALL SURFACES BETWEEN
ND THE ROOT DIA
ACHINE AREA F PER EPS 13066
.002 -.004 PER SURFACE

SURFACES AS REQUIRED

TE SURFACE E TO FINISH SIZE

MATERIAL- AMS 6265
STEEL FORGED BARS

BREAK SHARP EDGES .010 U0S

DIA A SHALL BE CONCENTRIC WITH PD WITHIN .002 TIR

MACHINE ALL OVER. PEEN GEAR TEETH PER EPS 12140
FOLLOWED BY EPS 12176
REMAINING SURFACES MAY BE PEENED PER EPS12140
UNLESS SPECIFICALLY CONTROLLED BY A √ SYMBOL

SURFACE CHARACTERISTICS NOT CONTROLLED BY A √
SYMBOL SHALL BE COMMENSURATE WITH GOOD MANU-
FACTURING PRACTICES WHICH PRODUCE ACCEPTABLE
QUALITY LEVELS.

HEAT TREAT PER EPS 202

CASE HARDEN GEAR TEETH OUTSIDE 1.570 DIA
(OPTIONAL TO CASE HARDEN ALL OVER) EFFECTIVE CASE
DEPTHS AS FOLLOWS:
   .020-.030 BEFORE FINISHING
   .015-.030 AFTER FINISHING
ROCKWELL HARDNESS – CASE C58 MIN
                     CORE C34 MIN

INSPECT PER EIS 985 (MAGNETIC)

BLACK OXIDE PER AMS 2485

ALL DIMENSIONS TO BE MET AFTER PROCESSING

FORGINGS SHALL CONFORM TO EDI 138 AND EIS 502

3779.

BREAK TIP .005 MAX
.015 -.030 ROUND BREAK
.050 R MIN
16/ SURFAC
40/ B
SHOT

ENLARGED VIEW OF GEAR PROFI
SCALE NONE

TOOTH SPACE SHALL BE FULLY GROUND INCLUDIN
DIA AND ADJACENT FILLETS AFTER HEAT
DISCONTINUITY SHALL OCCUR AT THE BLEND O
RADIUS WITH THE ROOT DIA AND INVOLUTE S
PEEN THE ENTIRE TOOTH SPACE PER EPS I2
BY EPS 12176 AFTER GRINDING, THEN NONE SURFA
SHOWN. REMAINING SURFACES MAY BE PEENED P
UNLESS SPECIFICALLY CONTRLLED BY A √ SYMBO
BY GRINDING TO BE UNIFORM ON PROFILE, FILLE
ROOT DIA WITHIN .002. MEASURE AND RECORD FI
DISTANCE OVER PINS AND ROOT DIA BEFORE AND

.502
.498

1.910
1.900
TYPICAL
12 PLACES

SIDE A

REMOVE 12 TEETH SPACED AS SHOWN AFTER
GEAR TEETH ARE TO FINISHED SIZE AND ALL
SIDE B GEAR TOOTH INSPECTION REQUIREMENTS
ARE COMPLETED

1.5005 DIA A
1.5000

CSK 90° TO 1.560 DIA
(BOTH SIDES)

ELECTRO CHEMICAL ETCH ALLISON PART NO AND
LAST. CHANGE LETTER, 'SER' AND SERIAL NO.
HERE PER AS 478-741 OR 742

ELECTRO CHEMICAL ETCH POSITION NUMBERS ON TEETH
AS SHOWN. RECORD INVOLUTE PROFILE AND LEAD
CHECKS FOR SIDE A OF TEETH 1,2,3, AND 4: AND FOR
SIDE B OF TEETH X1, X2, X3, AND X4. RECORD
POSITION NUMBERS ON TOOTH TO TOOTH SPACING
ERROR CHECKS

DIA A
BREA
MACH

SURFA
SYMBOL
FACTUR
QUALITY
HEAT
CASE
(OPTI
DEPT

ROCK

INSP

NITAL
BLAC

ALL D

FORG.

GEAR TOOTH CONTROL

| INVOLUTE PROFILE TOLERANCE | | |
|---|---|---|
| SIDE A | OD | SIDE B |
| 0.4° .0000 | OD8 | |
| 0.9 U -.0002 | | |
| 10.0° | PD | SAME |
| 23.3 U | | |
| 21.3° .0000 | AFCA | |
| 45.6 U -.0002 | | |
| 21.9° | AFD | |
| 47.2 U | | |
| 37.5° | BCD | |
| 60.8 U | | |

| SPACING TOLERANCE | |
|---|---|
| A .0002 | B .0002 |

| LEAD TOLERANCE | |
|---|---|
| A ±.0001 | B ±.0001 |

| FULLNESS TOLERANCE | |
|---|---|
| A .0000 | B .0000 |
| .0005 | .0005 |

MAXIMUM HOLLOW IN FORM .0001

| NOTE | 1 UNIT = .0147 IN. |
|---|---|

SPUR GEAR DATA
6 PITCH 24 TEETH
25° PRESSURE ANGLE
DISTANCE OVER TWO .2880 DIA PINS - 4.4012 +.0000/-.0039
ROOT DIA - 3.600 ±.002
PITCH DIA - 4.0000
OUTSIDE DIA - 4.3333 ±.0000/-.0000
ACTIVE PROFILE OUTSIDE 3.7571 DIA
GEAR TOOTH ELEMENTS SHALL BE IN ACCORDANCE WITH EDI 9
REFERENCE
ARC TOOTH THICKNESS AT PD - .2618 +.0000/-.0000
BASE CIRCLE DIA - 3.6252

MATERIAL - AMS 6265
STEEL FORGED BARS

Figure 119. Fatigue Test Gear Configuration 9—EX-78780.

165

A

BREAK TIP .005 MAX

.015 -.030 ROUND BREAK

.050 R MIN

SURFACE E

16/

40/

BEFORE
SHOT PEENING

**ENLARGED VIEW OF GEAR PROFILE**
**SCALE NONE**

TOOTH SPACE SHALL BE FULLY GROUND INCLUDING ROOT
DIA AND ADJACENT FILLETS AFTER HEAT TREAT. NO
DISCONTINUITY SHALL OCCUR AT THE BLEND OF THE FILLET
RADIUS WITH THE ROOT DIA AND INVOLUTE SURFACES. SHOT
PEEN THE ENTIRE TOOTH SPACE PER EPS 12140 FOLLOWED
BY EPS 12176 AFTER GRINDING, THEN HONE SURFACE E TO VALUE
SHOWN. REMAINING SURFACES MAY BE PEENED PER EPS 12140
UNLESS SPECIFICALLY CONTRLLED BY A √ SYMBOL. STOCK REMOVAL
BY GRINDING TO BE UNIFORM ON PROFILE, FILLET RADIUS AND
ROOT DIA WITHIN .002. MEASURE AND RECORD FILLET RADIUS,
DISTANCE OVER PINS AND ROOT DIA BEFORE AND AFTER GRINDING

REMOVE 12 TEETH SPACED AS SHOWN AFTER
GEAR TEETH ARE TO FINISHED SIZE AND ALL
SIDE B GEAR TOOTH INSPECTION REQUIREMENTS
ARE COMPLETED

DIA A SHALL BE CONCENTRIC WITH PD WITHIN .002 TIR

BREAK SHARP EDGES .010 UOS

MACHINE ALL OVER.

⊛

TRO CHEMICAL ETCH POSITION NUMBERS ON TEETH
OWN. RECORD INVOLUTE PROFILE AND LEAD
KS FOR SIDE A OF TEETH 1,2,3, AND 4: AND FOR
B OF TEETH X1, X2, X3, AND X4. RECORD
TION NUMBERS ON TOOTH TO TOOTH SPACING
OR CHECKS

SURFACE CHARACTERISTICS NOT CONTROLLED BY A √
SYMBOL SHALL BE COMMENSURATE WITH GOOD MANU-
FACTURING PRACTICES WHICH PRODUCE ACCEPTABLE
QUALITY LEVELS.

HEAT TREAT PER EPS 202

CASE HARDEN GEAR TEETH OUTSIDE 3.340 DIA
(OPTIONAL TO CASE HARDEN ALL OVER) EFFECTIVE CASE
DEPTHS AS FOLLOWS:
.035 -.045 BEFORE FINISHING
.030 -.045 AFTER FINISHING
ROCKWELL HARDNESS - CASE C58 MIN
CORE C34 MIN

INSPECT PER EIS 985 (MAGNETIC)

NITAL ETCH PER EIS 1510 THEN
BLACK OXIDE PER AMS 2485

MATERIAL - AMS 6265
STEEL FORGED BARS

.9000
.8039

ALL DIMENSIONS TO BE MET AFTER PROCESSING

FORGING TO CONFORM TO EDI 138 AND EIS 502

WITH EDI 9

BREAK TIP .005 MAX

.015 -.030 ROUNDED BREAK

$\frac{16}{}$ SURFACE $E$

.025 R MIN

$\frac{40}{}$ BEFORE SHOT PEENING

ENLARGED VIEW OF GEAR PROFILE
SCALE NONE

TOOTH SPACE SHALL BE FULLY GROUND INCLUDING ROOT DIA AND ADJACENT FILLETS AFTER HEAT TREAT. NO DISCONTINUITY SHALL OCCUR AT THE BLEND OF THE FILLET RADIUS WITH THE ROOT DIA AND INVOLUTE SURFACES. SHOT PEEN THE ENTIRE TOOTH SPACE PER EPS 12140 FOLLOWED BY EPS 12176 AFTER GRINDING, THEN HONE SURFACE E TO VALUE SHOWN. REMAINING SURFACES MAY BE PEENED PER EPS 12140 UNLESS SPECIFICALLY CONTROLLED BY A √ SYMBOL. STOCK REMOVAL BY GRINDING TO BE UNIFORM ON PROFILE, FILLET RADIUS AND ROOT DIA WITHIN .002. MEASURE AND RECORD FILLET RADIUS, DISTANCE OVER PINS AND ROOT DIA BEFORE AND AFTER GRINDING

.252
.248

.7505 DIA
.7500 A

$\frac{32}{}$

REMOVE 12 TEETH SPACED AS SHOWN AFTER GEAR TEETH ARE TO FINISHED SIZE AND ALL GEAR TOOTH INSPECTION REQUIREMENTS ARE COMPLETED

Ⓑ

.998 TYPICAL
.948 12 PLACES

SIDE B

SIDE A

CSK 90° TO .810 DIA
(BOTH SIDES)

ELECTRO CHEMICAL ETCH ALLISON PART NO. AND LAST CHANGE LETTER, 'SER' AND SERIAL NO. HERE PER AS 478-7A1- OR 742

ELECTRO CHEMICAL ETCH POSITION NUMBERS ON TEETH AS SHOWN. RECORD INVOLUTE PROFILE AND LEAD CHECKS FOR SIDE A OF TEETH 1, 2, 3 AND 4; AND FOR SIDE B OF TEETH X1, X2, X3 AND X4. RECORD POSITION NUMBERS ON TOOTH TO TOOTH SPACING ERROR CHECKS

DIA A

BREAK

MACHI

SURFACE
SYMBOL S
FACTURING
QUALITY

HEAT

CASE
(OPTIO
DEPTH

ROCKW

INSPE

| GEAR TOOTH CONTROL | | | |
|---|---|---|---|
| INVOLUTE PROFILE TOLERANCE | | | |
| SIDE A | OD | SIDE B | |
| 0.9° | .0000 | ODB | |
| 0.9 U | -.0002 | | |
| 10.8° | PD | SAME | |
| 11.6 U | | | |
| 20.7° | .0000 | AFCA | |
| 22.3 U | -.0002 | | |
| 22.3 U | | APD | |
| 23.9 U | | | |
| 37.5° | | BCD | |
| 40.4 U | | | |

| SPACING TOLERANCE | | |
|---|---|---|
| A .0002 | B .0002 | |

| LEAD TOLERANCE | | |
|---|---|---|
| A ±.0001 | B ±.0001 | |

| FULLNESS TOLERANCE | | |
|---|---|---|
| A .0000 | B .0000 | |

MAXIMUM HOLLOW IN FORM .0001

| NOTE | 1 UNIT = .0147 IN |
|---|---|

MATERIAL- AMS 6265
STEEL FORGED BARS

NITAL
BLACK

ALL DI

FORGIN

SPUR GEAR DATA
 12 PITCH 24 TEETH
 25° PRESSURE ANGLE
 DISTANCE OVER TWO .1440 DIA PINS= 2.2006 +.0000 -.0019
 ROOT DIA= 1.800 ±.002
 PITCH DIA= 2.0000
 OUTSIDE DIA= 2.1667 +.0000 -.0000
ACTIVE PROFILE OUTSIDE 1.8759 DIA
GEAR TOOTH ELEMENTS SHALL BE IN ACCORDANCE WITH EDI 9
REFERENCE
 ARC TOOTH THICKNESS AT PD= .1909 +.0000 -.0000
 BASE CIRCLE DIA= 1.8126

Figure 120. Fatigue Test Gear Configuration 10—EX-78781.
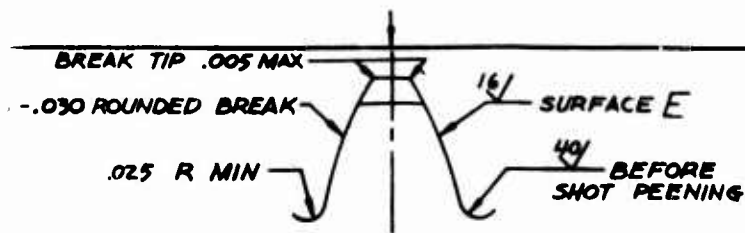
BREAK TIP .005 MAX
-.030 ROUNDED BREAK
.025 R MIN
SURFACE E
16/
40/ BEFORE
SHOT PEENING

ENLARGED VIEW OF GEAR PROFILE
SCALE NONE

)OTH SPACE SHALL BE FULLY GROUND INCLUDING ROOT
)A AND ADJACENT FILLETS AFTER HEAT TREAT. NO
)SCONTINUITY SHALL OCCUR AT THE BLEND OF THE FILLET
\DIUS WITH THE ROOT DIA AND INVOLUTE SURFACES. SHOT
>EN THE ENTIRE TOOTH SPACE PER EPS 12140 FOLLOWED
' EPS 12176 AFTER GRINDING, THEN HONE SURFACE E TO VALUE
/OWN. REMAINING SURFACES MAY BE PEENED PER EPS 12140 UNLESS
'ECIFICALLY CONTROLLED BY A √ SYMBOL. STOCK REMOVAL BY
RINDING TO BE UNIFORM ON PROFILE, FILLET RADIUS AND ROOT
\ WITHIN .002. MEASURE AND RECORD FILLET RADIUS, DISTANCE
FR PINS AND ROOT DIA BEFORE AND AFTER GRINDING

ONE 12 TEETH SPACED AS SHOWN
R GEAR TEETH ARE TO FINISHED
AND ALL GEAR TOOTH INSPECTION
IIREMENTS ARE COMPLETED
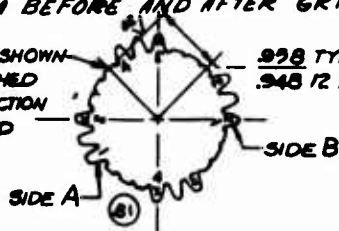
.998 TYPICAL
.948 12 PLACES

SIDE A
SIDE B

TRO CHEMICAL ETCH POSITION NUMBERS
!! TEETH AS SHOWN. RECORD INVOLUTE
)FILE AND LEAD CHECKS FOR SIDE A
TEETH 1, 2, 3, AND 4; AND FOR SIDE B
TEETH X1, X2, X3 AND X4. RECORD POSITION
\ MBERS ON TOOTH TO TOOTH SPACING
' ROR CHECKS

MATERIAL - AMS 6265
STEEL FORGED BARS

2.2006 +.0000 / -.0039

DIA
:ORDANCE WITH EDI 9
.0818

DIA A SHALL BE CONCENTRIC WITH PD WITHIN .002 TIR

BREAK SHARP EDGES .010 UOS

MACHINE ALL OVER.

SURFACE CHARACTERISTICS NOT CONTROLLED BY A √
SYMBOL SHALL BE COMMENSURATE WITH GOOD MANU-
FACTURING PRACTICES WHICH PRODUCE ACCEPTABLE
QUALITY LEVELS.

HEAT TREAT PER EPS 202

CASE HARDEN GEAR TEETH OUTSIDE 1.570 DIA
(OPTIONAL TO CASE HARDEN ALL OVER) EFFECTIVE CASE
DEPTHS AS FOLLOWS:
    .020-.030 BEFORE FINISHING
    .015-.030 AFTER FINISHING
ROCKWELL HARDNESS - CASE C58 MIN
                     CORE C34 MIN

INSPECT PER EIS 985 (MAGNETIC)

NITAL ETCH PER EIS 1510 THEN
BLACK OXIDE PER AMS 2485

ALL DIMENSIONS TO BE MET AFTER PROCESSING

FORGINGS SHALL CONFORM TO EDI 138 AND EIS 502

B

Figure 121. Fatigue Test Gear Configuration 11—EX-78782.

A

BREAK TIP .003 MAX

.015 - .030 ROUND BREAK — SURFACE E

.067 R MIN — BEFORE SHOT PEENING

### ENLARGED VIEW OF GEAR PROFILE
### SCALE NONE

TOOTH SPACE SHALL BE FULLY GROUND INCLUDING ROOT
DIA AND ADJACENT FILLETS AFTER HEAT TREAT. NO
DISCONTINUITY SHALL OCCUR AT THE BLEND OF THE FILLET
RADIUS WITH THE ROOT DIA AND INVOLUTE SURFACES. SHOT
PEEN THE ENTIRE TOOTH SPACE PER EPS 12140 FOLLOWED
BY EPS 12176 AFTER GRINDING, THEN HONE SURFACE E TO VALUE
SHOWN. REMAINING SURFACES MAY BE PEENED PER EPS 12140
UNLESS SPECIFICALLY CONTROLLED BY A √ SYMBOL. STOCK REMOVAL
BY GRINDING TO BE UNIFORM ON PROFILE, FILLET RADIUS AND
ROOT DIA WITHIN .002. MEASURE AND RECORD FILLET RADIUS,
DISTANCE OVER PINS AND ROOT DIA BEFORE AND AFTER GRINDING

REMOVE 12 TEETH SPACED AS SHOWN AFTER
GEAR TEETH ARE TO FINISHED SIZE AND ALL
GEAR TOOTH INSPECTION REQUIREMENTS
ARE COMPLETED

SIDE B

DIA A SHALL BE CONCENTRIC WITH PD WITHIN .002 TIR

BREAK SHARP EDGES .010 UOS

MACHINE ALL OVER.

CHEMICAL ETCH ALLISON PART NO. AND
CHANGE LETTER, 'SER' AND SERIAL NO.
PER AS 478-7A1 OR 7A2

SURFACE CHARACTERISTICS NOT CONTROLLED BY A √
SYMBOL SHALL BE COMMENSURATE WITH GOOD MANU-
FACTURING PRACTICES WHICH PRODUCE ACCEPTABLE
QUALITY LEVELS.

HEAT TREAT PER EPS 202

CASE HARDEN GEAR TEETH OUTSIDE 3.340 DIA
(OPTIONAL TO CASE HARDEN ALL OVER) EFFECTIVE CASE
DEPTHS AS FOLLOWS:
.035 - .045 BEFORE FINISHING
.030 - .045 AFTER FINISHING
ROCKWELL HARDNESS – CASE C58 MIN
CORE C34 MIN

MATERIAL- AMS 6265
STEEL FORGED BARS

INSPECT PER EIS 985 (MAGNETIC)

NITAL ETCH PER EIS 1510 THEN
BLACK OXIDE PER AMS 2485

ALL DIMENSIONS TO BE MET AFTER PROCESSING

$+.0000$
$-.0099$

FORGING SHALL CONFORM TO EDI 138 AND EIS 502

WITH EDI 9

1—EX-78782.

BREAK TIP .005 MAX

.015 - .030 ROUNDED BREAK

.033 R MIN

16/ SURFACE E

40/ BEFORE SHOT PEENING

ENLARGED VIEW OF GEAR PROFILE
SCALE NONE

TOOTH SPACE SHALL BE FULLY GROUND INCLUDING ROOT
DIA AND ADJACENT FILLETS AFTER HEAT TREAT. NO
DISCONTINUITY SHALL OCCUR AT THE BLEND OF THE FILLET
RADIUS WITH THE ROOT DIA AND INVOLUTE SURFACES. SHOT
PEEN THE ENTIRE TOOTH SPACE PER EPS 12140 FOLLOWED BY
EPS 12176 AFTER GRINDING, THEN HONE SURFACE E TO VALUE SHOW
REMAINING SURFACES MAY BE PEENED PER EPS 12140 UNLESS
SPECIFICALLY CONTROLLED BY A √ SYMBOL. STOCK REMOVAL BY
GRINDING TO BE UNIFORM ON PROFILE, FILLET RADIUS AND ROOT
DIA WITHIN .002. MEASURE AND RECORD FILLET RADIUS, DISTANCE
OVER PINS AND ROOT DIA BEFORE AND AFTER GRINDING.

.252
.248

32/

.7505
.7500 DIA A

958 TYPICAL
948 12 PLACES

SIDE A          SIDE B

(M)

CSK 90° TO .810 DIA
(BOTH SIDES)

ELECTRO CHEMICAL ETCH ALLISON PART NO.
AND LAST CHANGE LETTER "SEP" AND SERIAL NO.
HERE AS 478-7A1 OR 7A2

REMOVE 12 TEETH SPACED AS SHOWN
AFTER GEAR TEETH ARE TO FINISHED
SIZE AND ALL GEAR TOOTH INSPECTION
REQUIREMENTS ARE COMPLETED

(N) ELECTRO CHEMICAL ETCH POSITION NUMBERS
ON TEETH AS SHOWN. RECORD INVOLUTE
PROFILE AND LEAD CHECKS FOR SIDE A
OF TEETH 1,2,3, AND 4; AND FOR SIDE B
OF TEETH X1, X2, X3, AND X4. RECORD
POSITION NUMBERS ON TOOTH TO TOOTH
SPACING ERROR CHECKS

DIA A S

BREAK

MACHINE

SURFACE C/
SYMBOL SH
FACTURING
QUALITY LE

HEAT T

CASE H
(OPTION
DEPTHS
.C
.C
ROCKWE

INSPEC

NITAL
BLACK

ALL DIM
FORGING

MATERIAL- AMS 6265 STEEL FORGED BARS

SPUR GEAR DATA
12 PITCH 24 TEETH
25° PRESSURE ANGLE
DISTANCE OVER TWO .1440 DIA PINS- 2.2006 +.0000/-.0039
ROOT DIA- 1.800 ±.002
PITCH DIA- 2.0000
OUTSIDE DIA- 2.1667 +.0000/-.0035
ACTIVE PROFILE OUTSIDE 1.8759 DIA
GEAR TOOTH ELEMENTS SHALL BE IN ACCORDANCE WITH EDI 9
REFERENCE
ARC TOOTH THICKNESS AT PD- .1309 +.0000/-.0035
BASE CIRCLE DIA- 1.8126

| GEAR TOOTH CONTROL | | | |
|---|---|---|---|
| INVOLUTE PROFILE TOLERANCE | | | |
| SIDE A | OD | SIDE B | |
| 0.5°/0.5 U | .0000/-.0002 | 008 | |
| 10.8°/11.6 U | PD | SAME | |
| 20.7°/22.5 U | .0000/-.0002 | MFCA | |
| 31.3°/33.5 U | | AFD | |
| 37.5°/40.4 U | | BCD | |
| SPACING TOLERANCE | | | |
| A .0002 | | B .0003 | |
| LEAD TOLERANCE | | | |
| A ±.0001 | | B ±.0001 | |
| FULLNESS TOLERANCE | | | |
| A .0000/.0002 | | B .0000/.0002 | |
| MAXIMUM HOLLOW IN FORM .0001 | | | |
| NOTE | 1 UNIT = .0147 IN | | |

Figure 122. Fatigue Test Gear Configuration 12—EX-78783.

171

A

BREAK TIP .005 MAX

.030 ROUNDED BREAK

SURFACE E

.033 R MIN

BEFORE
SHOT PEENING

ENLARGED VIEW OF GEAR PROFILE
SCALE NONE

>TH SPACE SHALL BE FULLY GROUND INCLUDING ROOT
AND ADJACENT FILLETS AFTER HEAT TREAT. NO
CONTINUITY SHALL OCCUR AT THE BLEND OF THE FILLET
>IUS WITH THE ROOT DIA AND INVOLUTE SURFACES. SHOT
:N THE ENTIRE TOOTH SPACE PER EPS 12140 FOLLOWED BY
S 12176 AFTER GRINDING, THEN HONE SURFACE E TO VALUE SHOWN.
MAINING SURFACES MAY BE PEENED PER EPS 12140 UNLESS
CIFICALLY CONTROLLED BY A √ SYMBOL. STOCK REMOVAL BY
INDING TO BE UNIFORM ON PROFILE, FILLET RADIUS AND ROOT
WITHIN .002. MEASURE AND RECORD FILLET RADIUS, DISTANCE
R PINS AND ROOT DIA BEFORE AND AFTER GRINDING.

$\frac{.958}{.948}$ TYPICAL
12 PLACES

SIDE A

SIDE B

REMOVE 12 TEETH SPACED AS SHOWN
AFTER GEAR TEETH ARE TO FINISHED
SIZE AND ALL GEAR TOOTH INSPECTION
REQUIREMENTS ARE COMPLETED

ELECTRO CHEMICAL ETCH POSITION NUMBERS
ON TEETH AS SHOWN. RECORD INVOLUTE
PROFILE AND LEAD CHECKS FOR SIDE A
OF TEETH 1,2,3, AND 4; AND FOR SIDE B
OF TEETH X1, X2, X3, AND X4. RECORD
POSITION NUMBERS ON TOOTH TO TOOTH
SPACING ERROR CHECKS

MATERIAL- AMS 6265 STEEL FORGED BARS

$2006 ^{+.0000}_{-.0039}$

DIA A SHALL BE CONCENTRIC WITH PD WITHIN .002 TIR

BREAK SHARP EDGES .010 UOS

MACHINE ALL OVER.

SURFACE CHARACTERISTICS NOT CONTROLLED BY A √
SYMBOL SHALL BE COMMENSURATE WITH GOOD MANU-
FACTURING PRACTICES WHICH PRODUCE ACCEPTABLE
QUALITY LEVELS.

HEAT TREAT PER EPS 202

CASE HARDEN GEAR TEETH OUTSIDE 1.570 DIA
(OPTIONAL TO CASE HARDEN ALL OVER) EFFECTIVE CASE
DEPTHS AS FOLLOWS:
.020-.030 BEFORE FINISHING
.015-.030 AFTER FINISHING
ROCKWELL HARDNESS – CASE C 58 MIN
CORE C 34 MIN

INSPECT PER EIS 985 (MAGNETIC)

NITAL ETCH PER EIS 1510 THEN
BLACK OXIDE PER AMS 2485

ALL DIMENSIONS TO BE MET AFTER PROCESSING
FORGING SHALL CONFORM TO EDI 138 AND EIS 502

DANCE WITH EDI 9

78

BREAK TIP .003 MAX
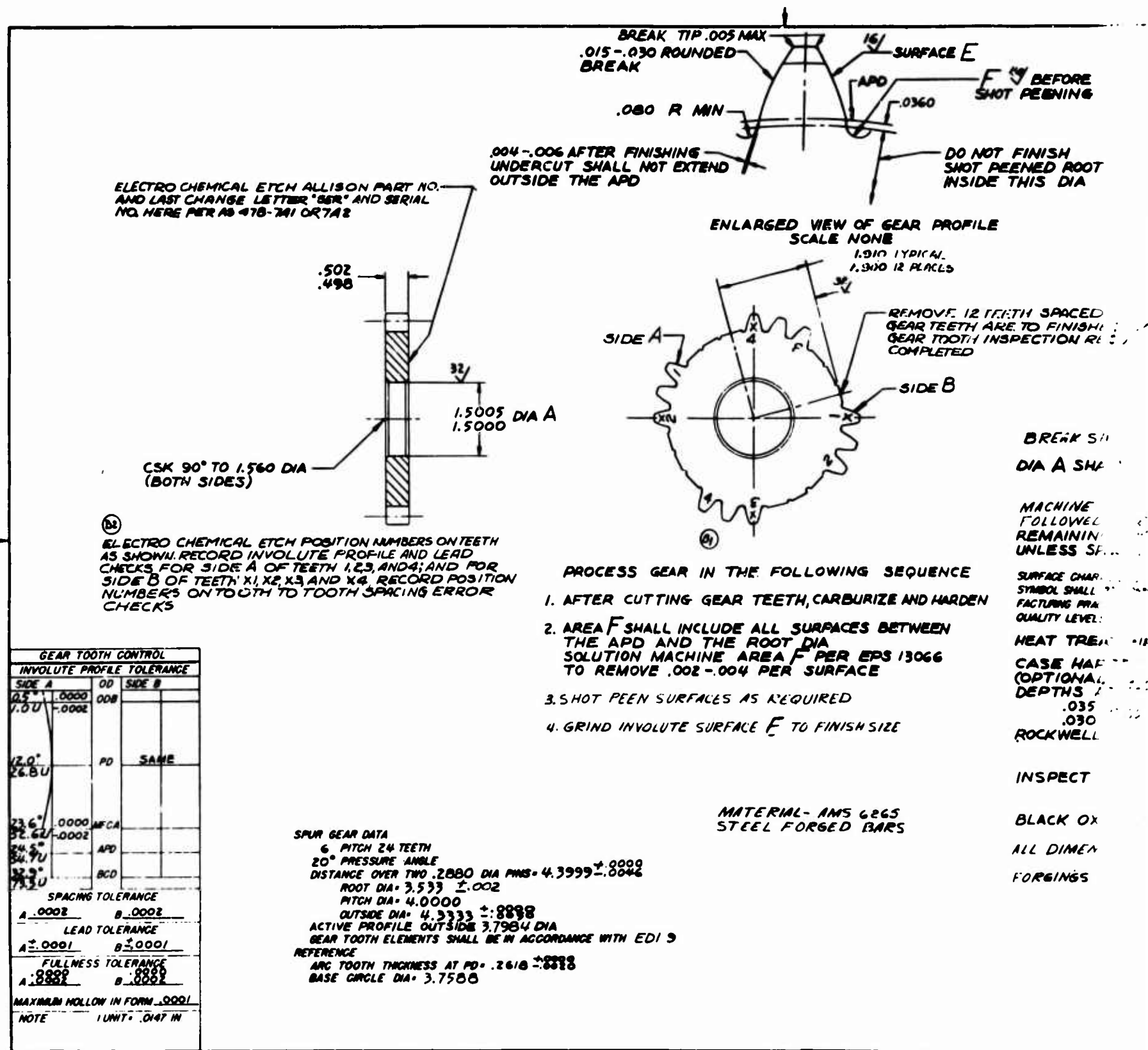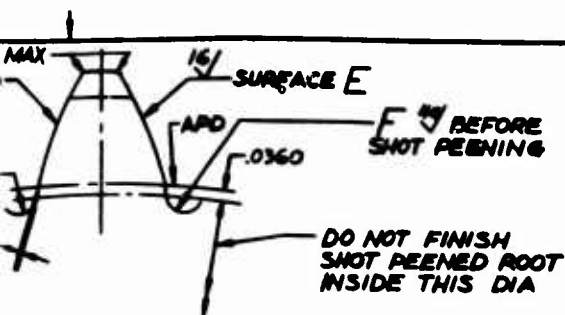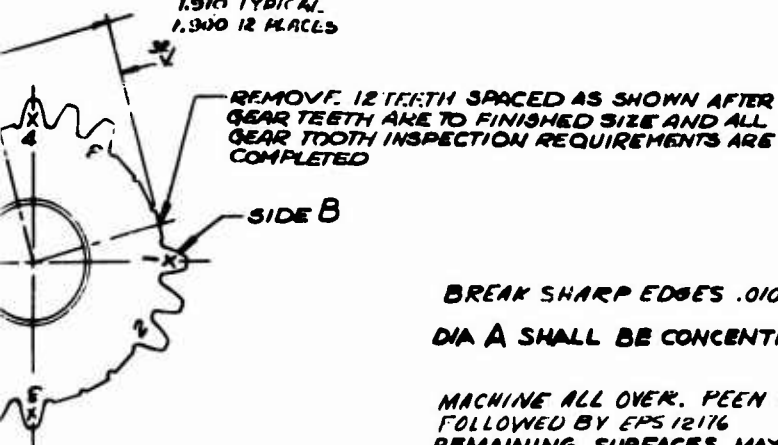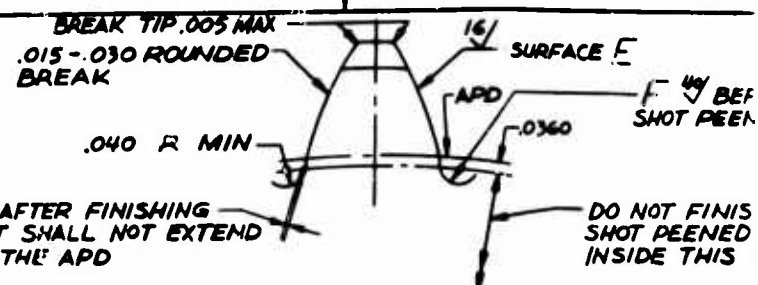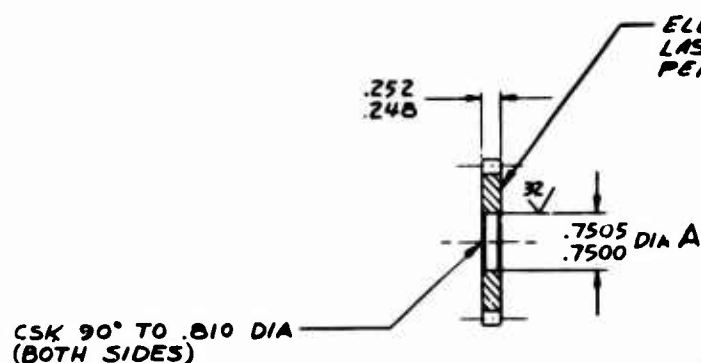.015-.030 ROUNDED BREAK
SURFACE E
F BEFO SHOT PEENII
.0360
.050 R MIN
.004-.006 AFTER FINISHING UNDERCUT SHALL NOT EXTEND OUTSIDE THE APD
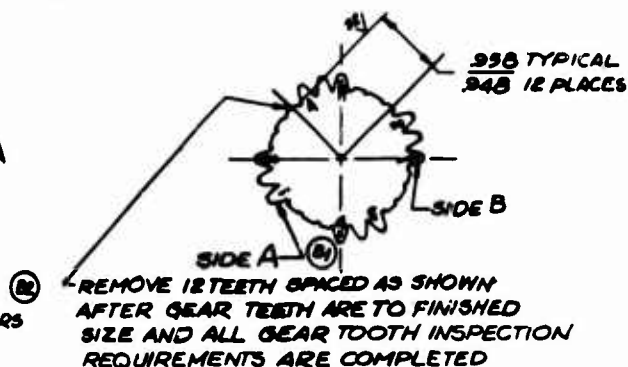DO NOT FINISH SHOT PEENED R INSIDE THIS D

ENLARGED VIEW OF GEAR PROFILE
SCALE NONE

ELECTRO CHEMICAL ETCH ALLISON PART NO. AND LAST CHANGE LETTER 'SER' AND SERIAL NO. HERE PER AS AS478-7A1 OR 7A2

.502
.498

.1910
.1900
TYPICAL
12 PLACES

SIDE A

REMOVE 12 TEETH SPACED AS SHOW GEAR TEETH ARE TO FINISHED SIZE GEAR TOOTH INSPECTION REQUIREM ARE COMPLETED

SIDE B

1.5005 DIA A
1.5000

CSK 90° TO 1.560 DIA
(BOTH SIDES)

ELECTRO CHEMICAL ETCH POSITION NUMBERS ON TEETH AS SHOWN. RECORD INVOLUTE PROFILE AND LEAD CHECKS FOR SIDE A OF TEETH 1,2,3, AND 4, AND FOR SIDE B OF TEETH X1, X2, X3, AND X4. RECORD POSITION NUMBERS ON TOOTH TO TOOTH SPACING ERROR CHECKS

PROCESS GEAR IN THE FOLLOWING SEQUENCE

1. AFTER CUTTING GEAR TEETH, CARBURIZE AND HARDEN

2. AREA F SHALL INCLUDE ALL SURFACES BETWEEN THE APD AND THE ROOT DIA SOLUTION MACHINE AREA F PER EPS 13066 TO REMOVE .002-.004 PER SURFACE

3. SHOT PEEN SURFACES AS REQUIRED

4. GRIND INVOLUTE SURFACE E TO FINISH SIZE

DIA A
BREA
MACHI
FOLLO
REMA
UNLES

SURFACE
SYMBOL
FACTUR
QUALITY

HEAT

CASE
(OPTIC
DEPTI

ROCKV

INSPE

BLACK

ALL D

FORG

**GEAR TOOTH CONTROL**

| INVOLUTE PROFILE TOLERANCE | | | |
|---|---|---|---|
| SIDE A | | OD | SIDE B |
| 0.4° | .0000 | OOB | |
| 0.5 U | -.0002 | | |
| 10.0° | | PD | SAME |
| 25.5 U | | | |
| 21.2° | .0000 | AFCA | |
| 45.6 U | -.0002 | | |
| 21.9° | | APD | |
| 47.2 U | | | |
| 32.5° | | BCD | |
| 54.8 U | | | |

SPACING TOLERANCE
A .0002    B .0002

LEAD TOLERANCE
A ±.0001   B ±.0001

FULLNESS TOLERANCE
A .0000    B .0000

MAXIMUM HOLLOW IN FORM .0001

NOTE    1 UNIT = .047 IN.

SPUR GEAR DATA
  6 PITCH 24 TEETH
  25° PRESSURE ANGLE
  DISTANCE OVER TWO .2880 DIA PINS= 4.4012 +.0000 -.0039
  ROOT DIA= 3.550 ±.002
  PITCH DIA= 4.0000
  OUTSIDE DIA= 4.3333 ±.0000
  ACTIVE PROFILE OUTSIDE 3.7571 DIA
  GEAR TOOTH ELEMENTS SHALL BE IN ACCORDANCE WITH EDI 9
REFERENCE
  ARC TOOTH THICKNESS AT PD= .2618
  BASE CIRCLE DIA= 3.6252

MATERIAL-AMS 6265 STEEL
FORGED BARS

Figure 123.   Fatigue Test Gear Configuration 13—EX-78784.

173

BREAK TIP .005 MAX

.015 -.030 ROUNDED BREAK

SURFACE E

F BEFORE SHOT PEENING

.050 R MIN-

.0360

.006 AFTER FINISHING
RCUT SHALL NOT EXTEND
IDE THE APD

DO NOT FINISH
SHOT PEENED ROOT
INSIDE THIS DIA

ENLARGED VIEW OF GEAR PROFILE
SCALE NONE

1.910 / 1.900
TYPICAL
12 PLACES

SIDE A

REMOVE 12 TEETH SPACED AS SHOWN AFTER
GEAR TEETH ARE TO FINISHED SIZE AND ALL
GEAR TOOTH INSPECTION REQUIREMENTS
ARE COMPLETED

SIDE B

A

DIA A SHALL BE CONCENTRIC WITH PD WITHIN .002 TIR

BREAK SHARP EDGES .010 UOS

MACHINE ALL OVER. PEEN GEAR TEETH PER EPS 12140
FOLLOWED BY EPS 12176
REMAINING SURFACES MAY BE PEENED PER EPS 12140
UNLESS SPECIFICALLY CONTROLLED BY A √ SYMBOL

SURFACE CHARACTERISTICS NOT CONTROLLED BY A √
SYMBOL SHALL BE COMMENSURATE WITH GOOD MANU-
FACTURING PRACTICES WHICH PRODUCE ACCEPTABLE
QUALITY LEVELS.

HEAT TREAT PER EPS 202

CASE HARDEN GEAR TEETH OUTSIDE 3.340 DIA
(OPTIONAL TO CASE HARDEN ALL OVER) EFFECTIVE CASE
DEPTHS AS FOLLOWS:
       .035 -.045 BEFORE FINISHING
       .030 -.045 AFTER FINISHING
ROCKWELL HARDNESS – CASE C58 MIN
                         CORE C34 MIN

INSPECT PER EIS 985 (MAGNETIC)

BLACK OXIDE PER AMS 2485

ALL DIMENSIONS TO BE MET AFTER PROCESSING

FORGING SHALL CONFORM TO EDI 138 AND EIS 502

PROCESS GEAR IN THE FOLLOWING SEQUENCE

1. AFTER CUTTING GEAR TEETH, CARBURIZE AND HARDEN

2. AREA F SHALL INCLUDE ALL SURFACES BETWEEN
   THE APD AND THE ROOT DIA
   SOLUTION MACHINE AREA F PER EPS 13066
   TO REMOVE .002 -.004 PER SURFACE

3 SHOT PEEN SURFACES AS REQUIRED

4 GRIND INVOLUTE SURFACE E TO FINISH SIZE

4.4012 +.0000 -.0039

MATERIAL-AMS 6265 STEEL
FORGED BARS

DIA
CORDANCE WITH EDI 9

BREAK TIP .005 MAX
.015-.030 ROUNDED
BREAK
16/ SURFACE E
APD
F B
SHOT PE

.025 R MIN
.0360

.004-.006 AFTER FINISHING
UNDERCUT SHALL NOT EXTEND
OUTSIDE THE APD

DO NOT FIN
SHOT PEENE
INSIDE THIS

ENLARGED VIEW OF GEAR PROFILE
SCALE NONE

ELECTRO CHEMICAL ETCH ALLISON PART NO. AND LAST
CHANGE LETTER, "SER" AND SERIAL NO. HERE PER A5478-7A1 OR 7A2

.252
.248

.958 TYPICAL
.948 12 PLACES

3R

.7505 DIA A
1.7500

SIDE B

SIDE A

CSK 90° TO .810 DIA
(BOTH SIDES)

B1

DIA
BRO
MAO
FOL
REN
UNL

REMOVE 12 TEETH SPACED AS
SHOWN AFTER GEAR TEETH ARE TO
FINISHED SIZE AND ALL GEAR
TOOTH INSPECTION REQUIREMENTS
ARE COMPLETED

PROCESS GEAR IN THE FOLLOWING SEQUENCE

SURF
SYMB
FACT
QUAL

ELECTRO CHEMICAL ETCH POSITION
NUMBERS ON TEETH AS SHOWN. RECORD
INVOLUTE PROFILE AND LEAD CHECKS FOR
SIDE A OF TEETH 1,2,3, AND 4; AND FOR
SIDE B OF TEETH X1, X2, X3, AND X4.
RECORD POSITION NUMBERS ON
TOOTH TO TOOTH SPACING ERROR
CHECKS

1. AFTER CUTTING GEAR TEETH, CARBURIZE AND HARDEN

2. AREA F SHALL INCLUDE ALL SURFACES BETWEEN
THE APD AND THE ROOT DIA
SOLUTION MACHINE AREA F PER EPS 13066
TO REMOVE .002-.004 PER SURFACE

3 SHOT PEEN SURFACES AS REQUIRED

4 GRIND INVOLUTE SURFACE E TO FINISH SIZE

HEA
CAS
(OP
DEF

ROC

INS

BLA

| GEAR TOOTH CONTROL | | | | |
|---|---|---|---|---|
| INVOLUTE PROFILE TOLERANCE | | | | |
| SIDE A | | OD | SIDE B | |
| 0.5° | .0000 | ODB | | |
| 0.9 U | -.0002 | | | |
| | | | | |
| 14.8° | | PD | SAME | |
| 11.6 U | | | | |
| | | | | |
| 20.7° | .0000 | MFCA | | |
| 22.3 U | -.0002 | | | |
| 22.3° | | A/D | | |
| 23.5 U | | | | |
| 37.5° | | BCD | | |
| 43.4 U | | | | |

ALL
FOR

SPACING TOLERANCE
A .0002          B .0002

LEAD TOLERANCE
A ±.0001        B ±.0001

FULLNESS TOLERANCE
A .0002          B .0002

MAXIMUM HOLLOW IN FORM .0001

NOTE      1 UNIT = .0147 IN.

SPUR GEAR DATA
12 PITCH 24 TEETH
25° PRESSURE ANGLE
DISTANCE OVER TWO .1440 DIA PINS = 2.2006 +.0000 -.0039
ROOT DIA = 1.775 ±.002
PITCH DIA = 2.0000
OUTSIDE DIA = 2.1667 +.0000 -.0000
ACTIVE PROFILE OUTSIDE 1.8759 DIA
GEAR TOOTH ELEMENTS SHALL BE IN ACCORDANCE WITH EDI 9
REFERENCE
ARC TOOTH THICKNESS AT PD = .1309 +.0000 -.0000
BASE CIRCLE DIA = 1.8126

MATERIAL-AMS 6265 STEEL
FORGED BARS

Figure 124.   Fatigue Test Gear Configuration 14—EX-78785.

A

BREAK TIP .005 MAX

.015 - .030 ROUNDED
BREAK

16/ SURFACE E

APD

F √ BEFORE
SHOT PEENING

.025 R MIN

.0360

.006 AFTER FINISHING
RCUT SHALL NOT EXTEND
IDE THE APD

DO NOT FINISH
SHOT PEENED ROOT
INSIDE THIS DIA

ENLARGED VIEW OF GEAR PROFILE
SCALE NONE

ECTRO CHEMICAL ETCH ALLISON PART NO. AND LAST
ANGE LETTER, "SER" AND SERIAL NO. HERE PER A3478-7A1 OR 7A2

.958 TYPICAL
.948 12 PLACES

SIDE A

SIDE B

DIA A SHALL BE CONCENTRIC WITH PD WITHIN .002 TIR

BREAK SHARP EDGES .010 UOS

MACHINE ALL OVER. PEEN GEAR TEETH PER EPS 12140
FOLLOWED BY EPS 12176
REMAINING SURFACES MAY BE PEENED PER EPS 12140
UNLESS SPECIFICALLY CONTROLLED BY A √ SYMBOL

SURFACE CHARACTERISTICS NOT CONTROLLED BY A √
SYMBOL SHALL BE COMMENSURATE WITH GOOD MANU-
FACTURING PRACTICES WHICH PRODUCE ACCEPTABLE
QUALITY LEVELS.

HEAT TREAT PER EPS 202

CASE HARDEN GEAR TEETH OUTSIDE 1.570 DIA
(OPTIONAL TO CASE HARDEN ALL OVER) EFFECTIVE CASE
DEPTHS AS FOLLOWS:
.020-.030 BEFORE FINISHING
.015-.030 AFTER FINISHING
ROCKWELL HARDNESS — CASE C58 MIN
CORE C34 MIN
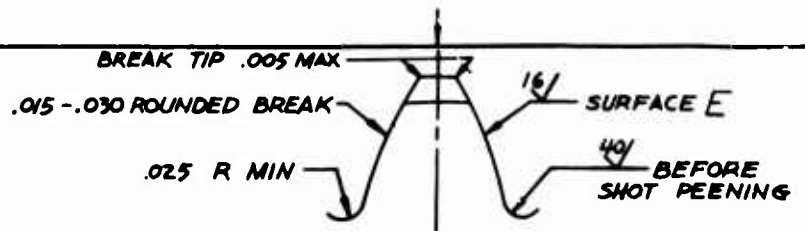
INSPECT PER EIS 985 (MAGNETIC)

BLACK OXIDE PER AMS 2485

ALL DIMENSIONS TO BE MET AFTER PROCESSING

FORGING SHALL CONFORM TO EDI 138 AND EIS 502

PROCESS GEAR IN THE FOLLOWING SEQUENCE

1. AFTER CUTTING GEAR TEETH, CARBURIZE AND HARDEN

2. AREA F SHALL INCLUDE ALL SURFACES BETWEEN
   THE APD AND THE ROOT DIA
OR SOLUTION MACHINE AREA F PER EPS 13066
   TO REMOVE .002-.004 PER SURFACE

3. SHOT PEEN SURFACES AS REQUIRED

4. GRIND INVOLUTE SURFACE E TO FINISH SIZE

2.2006 +.0000 -.0039

MATERIAL - AMS 6265 STEEL
FORGED BARS

DIA
:CORDANCE WITH EDI 9

±.0000

B

ELECTRO CHEMICAL ETCH ALLISON PART NO. AND LAST CHANGE LETTER, "SER" AND SERIAL NO. HERE PER AS 478-7A1 OR 7A2

BREAK TIP .005 MAX
.015-.030 ROUNDED BREAK
.067 R MIN
SURFACE E
APD
.0360
SHOT
.004-.006 AFTER FINISHING
UNDERCUT SHALL NOT EXTEND OUTSIDE THE APD
DO NOT FIL:
SHOT PEE!:L
INSIDE T

ENLARGED VIEW OF GEAR PROFILE
SCALE NONE

.502
.498

32/

1.5005 DIA A
1.5000

CSK 90° TO 1.560 DIA
(BOTH SIDES)

1.910 TYPICAL
1.900 12 PLACES
REMOVE 12 TE
GEAR TEETH
ALL GEAR TO
ARE COMPL.
SIDE A
SIDE B

PROCESS GEAR IN THE FOLLOWING SEQUENCE

1. AFTER CUTTING GEAR TEETH, CARBURIZE AND HARDEN

2. AREA F SHALL INCLUDE ALL SURFACES BETWEEN THE APD AND THE ROOT DIA
   SOLUTION MACHINE AREA F PER EPS 13066 TO REMOVE .002-.004 PER SURFACE

3. SHOT PEEN SURFACES AS REQUIRED

4. GRIND INVOLUTE SURFACE E TO FINISH SIZE

(82) ELECTRO CHEMICAL ETCH POSITION NUMBERS ON TEETH AS SHOWN. RECORD INVOLUTE PROFILE AND LEAD CHECKS FOR SIDE A OF TEETH 12, 3, AND 4; AND FOR SIDE B OF TEETH X1, X2, X3, AND X4. RECORD POSITION NUMBERS ON TOOTH TO TOOTH SPACING ERROR CHECKS

GEAR TOOTH CONTROL

| INVOLUTE PROFILE TOLERANCE | | | |
|---|---|---|---|
| SIDE A | OD | SIDE B | |
| 0.4° | .0000 | ODB | |
| 0.9 U | -.0002 | | |
| 10.8° | | PD | SAME |
| 25.3 U | | | |
| 21.2° | .0000 | MFCA | |
| 45.6 U | -.0002 | | |
| 21.9° | | APD | |
| 47.2 U | | | |
| 37.5° | | BCD | |
| 80.8 U | | | |

SPACING TOLERANCE
A .0002    B .0002

LEAD TOLERANCE
A ±.0001   B ±.0001

FULLNESS TOLERANCE
A .0000    B .0000

MAXIMUM HOLLOW IN FORM .0001

NOTE        1 UNIT = .0147 IN.

SPUR GEAR DATA
6 PITCH 24 TEETH
25° PRESSURE ANGLE
DISTANCE OVER TWO .2880 DIA PINS = 4.4012    +.0000 -.0009
ROOT DIA = 3.550 ±.002
PITCH DIA = 4.0000
OUTSIDE DIA = 4.3333 ±.0000
ACTIVE PROFILE OUTSIDE 3.7571 DIA
GEAR TOOTH ELEMENTS SHALL BE IN ACCORDANCE WITH EDI 9
REFERENCE
ARC TOOTH THICKNESS AT PD = .2618 ±.0000
BASE CIRCLE DIA = 3.6252

**Figure 125.  Fatigue Test Gear Configuration 15—EX-78786.**

A

BREAK TIP .005 MAX
.015 - .030 ROUNDED BREAK

16/ SURFACE E

APD

F 4/ BEFORE SHOT PEENING

.067 R MIN

.0360

AFTER FINISHING AREA SHALL NOT EXTEND ..D : THE APD

DO NOT FINISH SHOT PEENED ROOT INSIDE THIS DIA

ENLARGED VIEW OF GEAR PROFILE
SCALE NONE

1.910 TYPICAL
1.900 12 PLACES

REMOVE 12 TEETH SPACED AS SHOWN AFTER GEAR TEETH ARE TO FINISHED SIZE AND ALL GEAR TOOTH INSPECTION REQUIREMENTS ARE COMPLETED

SIDE A

SIDE B
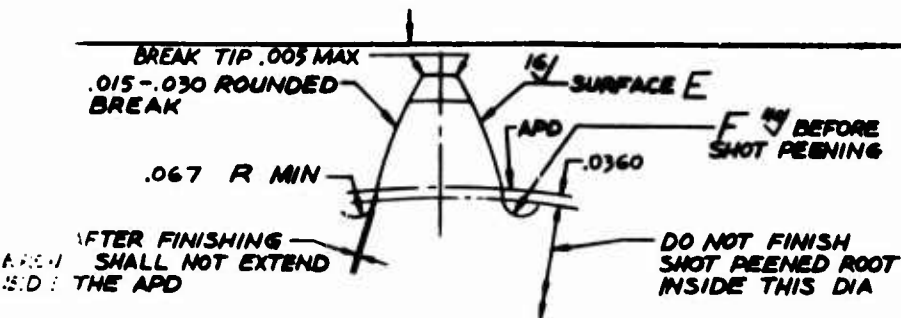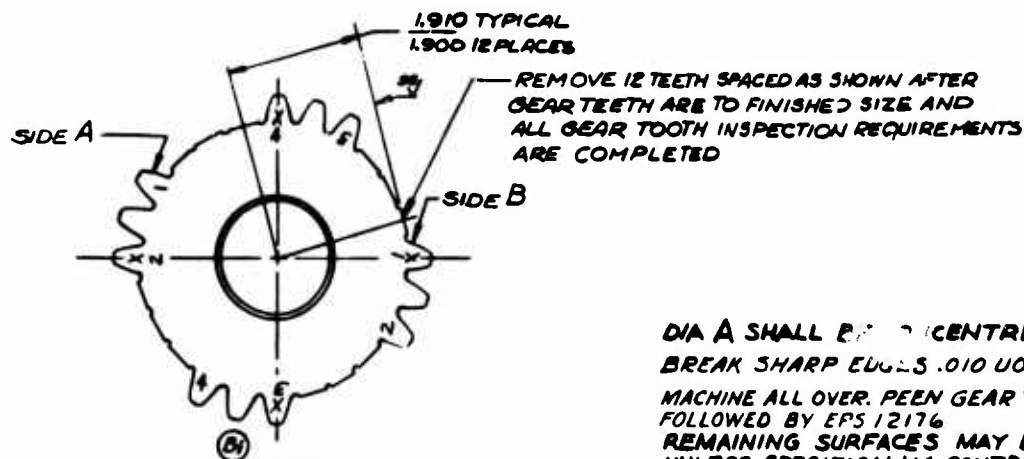
PROCESS GEAR IN THE FOLLOWING SEQUENCE

AFTER CUTTING GEAR TEETH, CARBURIZE AND HARDEN

AREA F SHALL INCLUDE ALL SURFACES BETWEEN THE APD AND THE ROOT DIA
SOLUTION MACHINE AREA F PER EPS 13066 TO REMOVE .002 -.004 PER SURFACE

SHOT PEEN SURFACES AS REQUIRED

GRIND INVOLUTE SURFACE E TO FINISH SIZE

ELECTRO CHEMICAL ETCH POSITION NUMBERS ON TEETH AS SHOWN, RECORD INVOLUTE PROFILE AND LEAD CHECKS FOR SIDE A OF TEETH 1,2,3, AND 4; AND FOR SIDE B OF TEETH X1, X2, X3, AND X4. RECORD POSITION NUMBERS ON TOOTH TO TOOTH SPACING ERROR CHECKS

+.0000
-.0030
.012

DIA A SHALL BE CONCENTRIC WITH PD WITHIN .002 TIR

BREAK SHARP EDGES .010 UOS

MACHINE ALL OVER. PEEN GEAR TEETH PER EPS 12140 FOLLOWED BY EPS 12176
REMAINING SURFACES MAY BE PEENED PER EPS 12140 UNLESS SPECIFICALLY CONTROLLED BY A √ SYMBOL

SURFACE CHARACTERISTICS NOT CONTROLLED BY A √ SYMBOL SHALL BE COMMENSURATE WITH GOOD MANU- FACTURING PRACTICES WHICH PRODUCE ACCEPTABLE QUALITY LEVELS.

HEAT TREAT PER EPS 202

CASE HARDEN GEAR TEETH OUTSIDE 3.340 DIA (OPTIONAL TO CASE HARDEN ALL OVER) EFFECTIVE CASE DEPTHS AS FOLLOWS:
.035 -.045 BEFORE FINISHING
.030 -.045 AFTER FINISHING
ROCKWELL HARDNESS - CASE C58 MIN
CORE C34 MIN

INSPECT PER EIS 985 (MAGNETIC)

BLACK OXIDE PER AMS 2485

ALL DIMENSIONS TO BE MET AFTER PROCESSING

FORGING SHALL CONFORM TO EDI 138 AND EIS 502
MATERIAL: AMS 6265 STEEL
FORGED BARS

ANCE WITH EDI 9

BREAK TIP .005 MAX

.015 - .030 ROUNDED BREAK

SURFACE E

F ⁴⁰/ BEFOR SHOT PEENIN

.033 R MIN

APD

.0360

.004 - .006 AFTER FINISHING UNDERCUT SHALL NOT EXTEND OUTSIDE THE APD

DO NOT FINISH SHOT PEENED R INSIDE THIS DI

ENLARGED VIEW OF GEAR PROFILE
SCALE NONE

ELECTROCHEMICAL ETCH ALLISON PART NO. AND LAST CHANGE LETTER, "SER" AND SERIAL NO. HERE PER AS 478-7A1 OR 7A2

.252
.248

.958 TYPICAL
.948 12 PLACES

32

.7505 DIA A
.7500

CSK 90° TO .810 DIA
(BOTH SIDES)

SIDE B

SIDE A

BREA·

DIA A

REMOVE 12 TEETH SPACED AS SHOWN AFTER GEAR TEETH ARE TO FINISHED SIZE AND ALL GEAR TOOTH INSPECTION REQUIREMENTS ARE COMPLETED

ELECTRO CHEMICAL ETCH POSITION NUMBERS ON TEETH AS SHOWN, RECORD INVOLUTE PROFILE AND LEAD CHECKS FOR SIDE A OF TEETH 1,2 AND 4; AND FOR SIDE B OF TEETH X1, X2, X3 AND X4. RECORD POSITION NUMBERS ON TOOTH TO TOOTH SPACING ERROR CHECKS

MACHI
FOLLO
REMAI
UNLESS

PROCESS GEAR IN THE FOLLOWING SEQUENCE

SURFACE
SYMBOL S
FACTURIN
QUALITY (

1. AFTER CUTTING GEAR TEETH, CARBURIZE AND HARDEN

2. AREA F SHALL INCLUDE ALL SURFACES BETWEEN THE APD AND THE ROOT DIA SOLUTION MACHINE AREA F PER EPS 13066 TO REMOVE .002 - .004 PER SURFACE

HEAT

CASE
(OPTIC
DEPTH

3. SHOT PEEN SURFACES AS REQUIRED

4 GRIND INVOLUTE SURFACE E TO FINISH SIZE

ROCKW

INSPE

| GEAR TOOTH CONTROL | | |
| --- | --- | --- |
| INVOLUTE PROFILE TOLERANCE | | |
| SIDE A | OD | SIDE B |
| 0.9° 0.9 0 | .0000 -.0002 | ODA |
| 10.8° 11.6 0 | PD | SAME |
| 20.7° 22.3 0 | .0000 -.0002 | APCA |
| 22.3 1 23.9 0 | | APD |
| 37.5° 40.4 0 | | BCC |

MATERIAL - AMS 6265
STEEL FORGED BARS

BLACK

SPACING TOLERANCE
A .0002      B .0002

LEAD TOLERANCE
A ± .0001    B ± .0001

FULLNESS TOLERANCE
A .0000 .0005    B .0000 .0005

MAXIMUM HOLLOW IN FORM .0001

NOTE      1 UNIT = .0147 IN

SPUR GEAR DATA
12 PITCH 24 TEETH
25° PRESSURE ANGLE
DISTANCE OVER TWO .1440 DIA PINS = 2.2006 +.0000 -.0039
ROOT DIA = 1.775 ± .002
PITCH DIA = 2.0000
OUTSIDE DIA = 2.1667 +.0000 -.0000
ACTIVE PROFILE OUTSIDE 1.8759 DIA
GEAR TOOTH ELEMENTS SHALL BE IN ACCORDANCE WITH ED 9
REFERENCE
ARC TOOTH THICKNESS AT PD = .1309 +.0000 -.0018
BASE CIRCLE DIA = 1.8126

ALL DI

FORGIN

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Figure 126.  Fatigue Test Gear Configuration 16—EX-78787.

A

BREAK TIP .005 MAX

.015 - .030 ROUNDED BREAK

16/

SURFACE E

APD

F ⁴⁹/ BEFORE SHOT PEENING

.033 R MIN

.0360

.006 AFTER FINISHING
RCUT SHALL NOT EXTEND
IDE THE APD

DO NOT FINISH SHOT PEENED ROOT INSIDE THIS DIA

ENLARGED VIEW OF GEAR PROFILE
SCALE NONE

CTROCHEMICAL ETCH ALLISON PART NO.
LAST CHANGE LETTER, 'SER' AND SERIAL
HERE PER AS 478-7A1 OR 7A2

.958 TYPICAL
.948 12 PLACES

SIDE B

SIDE A

REMOVE 12 TEETH SPACED AS SHOWN
AFTER GEAR TEETH ARE TO FINISHED
SIZE AND ALL GEAR TOOTH INSPECTION
REQUIREMENTS ARE COMPLETED

PROCESS GEAR IN THE FOLLOWING SEQUENCE

1. AFTER CUTTING GEAR TEETH, CARBURIZE AND HARDEN

2. AREA F SHALL INCLUDE ALL SURFACES BETWEEN
THE APD AND THE ROOT DIA
SOLUTION MACHINE AREA F PER EPS 13066
TO REMOVE .002 -.004 PER SURFACE

3. SHOT PEEN SURFACES AS REQUIRED

4. GRIND INVOLUTE SURFACE E TO FINISH SIZE

MATERIAL - AMS 6265
STEEL FORGED BARS

2.2006 +.0000 -.0039

DIA
CORDANCE WITH EDI 9

+.0000 -.0038

BREAK SHARP EDGES .010 UOS

DIA A SHALL BE CONCENTRIC WITH PD WITHIN .002 TIR

MACHINE ALL OVER. PEEN GEAR TEETH PER EPS 12140
FOLLOWED BY EPS 12176
REMAINING SURFACES MAY BE PEENED PER EPS 12140
UNLESS SPECIFICALLY CONTROLLED BY A √ SYMBOL

SURFACE CHARACTERISTICS NOT CONTROLLED BY A √
SYMBOL SHALL BE COMMENSURATE WITH GOOD MANU-
FACTURING PRACTICES WHICH PRODUCE ACCEPTABLE
QUALITY LEVELS.

HEAT TREAT PER EPS 202

CASE HARDEN GEAR TEETH OUTSIDE 1.570 DIA
(OPTIONAL TO CASE HARDEN ALL OVER) EFFECTIVE CASE
DEPTHS AS FOLLOWS:
.020-.030 BEFORE FINISHING
.015-.030 AFTER FINISHING
ROCKWELL HARDNESS - CASE C 58 MIN
CORE C 34 MIN

INSPECT PER EIS 985 (MAGNETIC)

BLACK OXIDE PER AMS 2485

ALL DIMENSIONS TO BE MET AFTER PROCESSING

FORGINGS SHALL CONFORM TO EDI 138 AND EIS 502

Figure 127.   Main Accessory Drive Spur Gear (6829396).
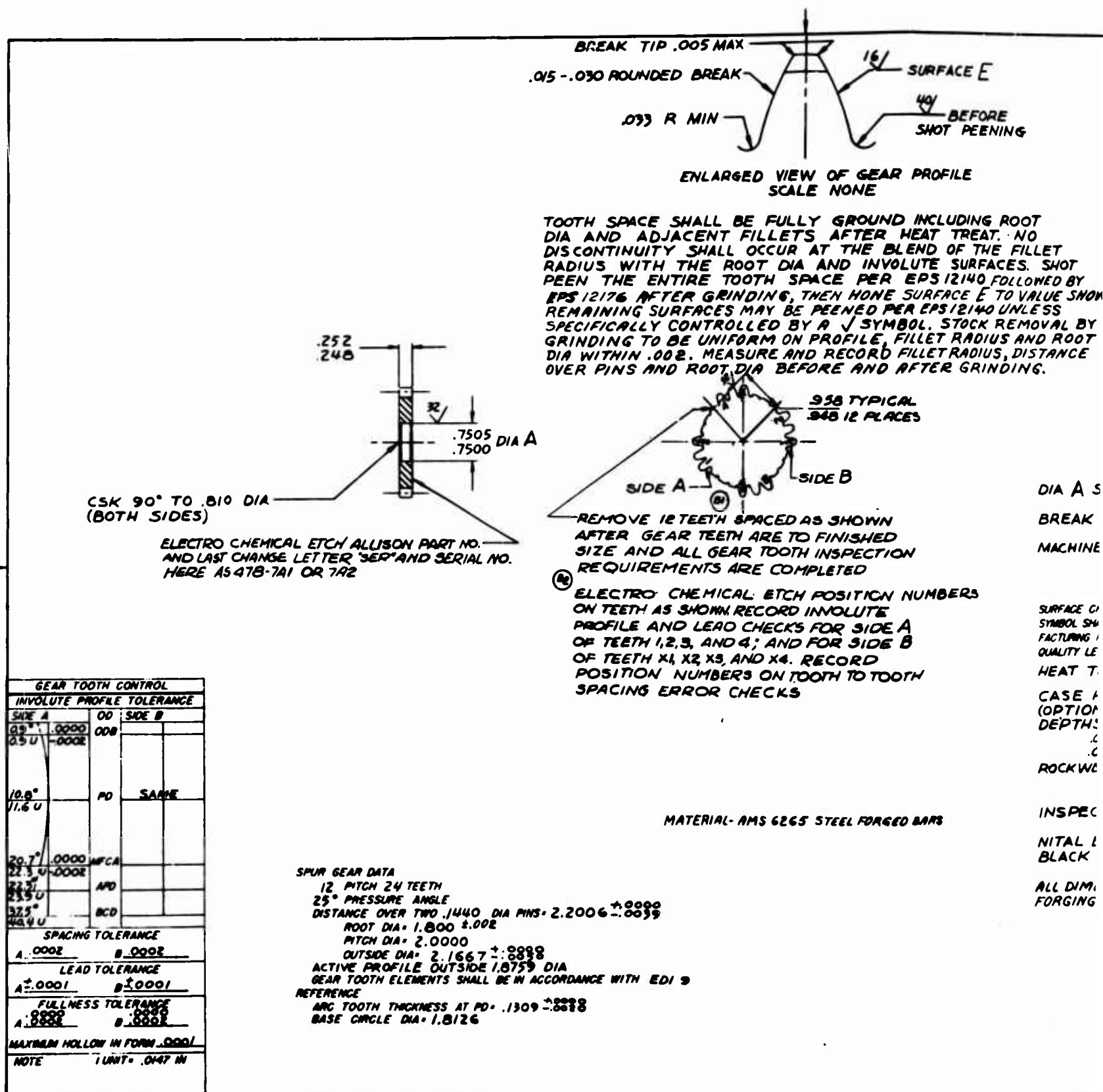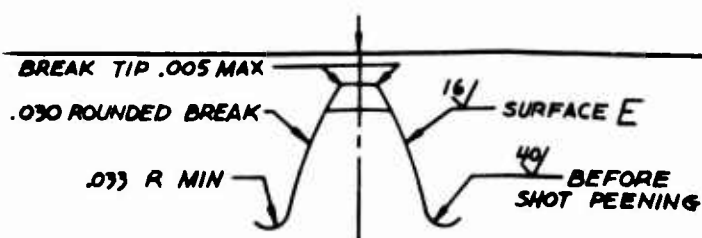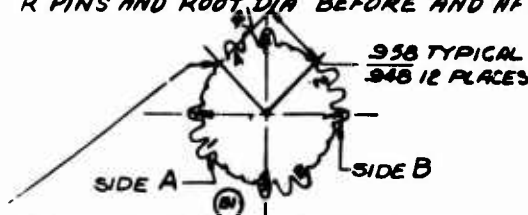
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CASE HARDEN ALL SURFACES OUTSIDE 6.34 DIA
EFFECTIVE CASE DEPTH TO BE .025-.035 EXCEPT SURFACES
FINISHED AFTER HARDENING TO BE .020-.035

SIDE A

ELECTROLYTIC ETCH "O" HERE FOR
OFFSET HOLE PER EPS 15400- OPPOSITE SIDE.



6.550 RD

7.490 OD

6.8000 PD

1.7500

B

1.8427 REF

CASE HARDEN THESE SURFACES
EFFECTIVE CASE DEPTH TO BE .025-.035 EXCEPT
SURFACES FINISHED AFTER HARDENING
TO BE .020-.035

SILVER PLATE PER AMS 2412
PLATE TO BE .0001-.0003 THICK

4.469 / 4.406 DIA

E  F

A — A

4.469 / 4.406 DIA

6.9370 / 2.9360 DIA

D

CHAMFER .05 X 45°

DO NOT BREAK THIS EDGE

5.750 DIA — H

6.027 / 6.013 DIA — G

ENLARGED VIEW 4 X SIZE
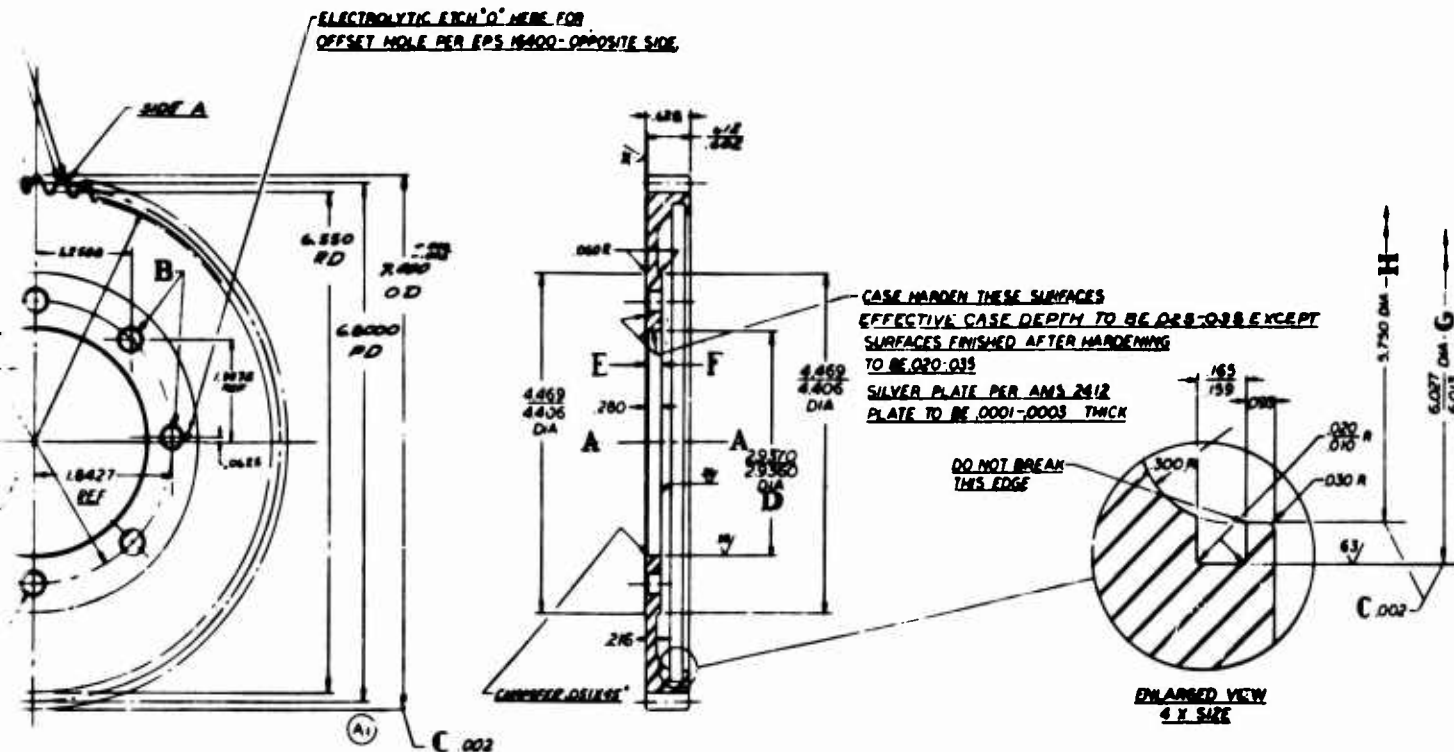
THE TOTAL ACCUMULATED SPACING ERROR
Ⓐ RELATIVE TO AXIS /A-A/ (INCLUDING PITCH LINE
ECCENTRICITY) SHALL NOT EXCEED .002 ON
SIDE 'A' & .004 ON SIDE 'B'

BREAK SHARP EDGES .010 UOS
CASE HARDEN WHERE SHOWN
ROCKWELL HARDNESS – CASE C 60 MIN (OR EQUIVALENT)
CORE C 30 MIN
FINISH DIA G AND H AFTER SHOT PEENING
AXIS A-A IS ESTABLISHED BY DIA D AND SURFACE E
FEATURES SHALL BE CONCENTRIC ABOUT AXIS A-A WITHIN
THE FIR SPECIFIED BY C
SURFACE F SHALL BE PARALLEL WITH SURFACE E WITHIN .002 FIR.
SHALL CONFORM TO EDI 138-1
FORGINGS SHALL CONFORM TO EIS 502
MACHINE ALL OVER. PEEN GEAR WEB & ADJACENT FILLETS
TO EPS 12120 (BEFORE PLATING)
ALL DIMENSIONS TO BE MET AFTER PLATING

...M DEFINED BY THIS DRAWING AND IDENTIFIED
...ON PART NUMBER HAS BEEN TESTED AND APPROVED
...OR USE IN ALLISON AIRCRAFT ENGINES
...IST OR PARTS CATALOG FOR SPECIFIC MODELS)
...F ITEM OR ONE FROM ANOTHER SOURCE SHALL NOT
...OUT PRIOR TEST AND APPROVAL BY ALLISON

...NG SOURCE APPROVAL REQ'D
...JG, ALSO FOR FINISHED PART
...TON AND APPROVAL BY ALLISON NO CHANGE SHALL
...MANUFACTURING PROCESSES OR MANUFACTURING
...WILL AFFECT THE RELIABILITY OF THIS PART

...ACTERISTICS NOT CONTROLLED BY A ✓
...BE COMMENSURATE WITH GOOD MANU-
...TICES WHICH PRODUCE ACCEPTABLE
...S

ur Gear (6829396).

B

Figure 128. Propeller Brake Outer Member (6829395).

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A

OPTIONAL TO PLATE THIS RADIUS
AND THIS SURFACE

PLATE INSIDE SURFACES .0001-.0003
THICK FOR THIS DISTANCE PER AMS 2406

.015-.030 ROUNDED BREAK
(BOTH SIDES)

**B** .001

DO NOT BREAK THESE EDGES
CHAMFER .047 x 45°

**B** .005

SIDE A — — SIDE B

SEVEN SLOTS LOCATED AS
SHOWN AND EQUALLY SPACED
WITHIN .005 OF TRUE POSITION

BREAK SHARP EDGE .015-.030

THIS DIA TO THIS DIMENSION
PRIOR TO PLATING
GAGE LINE

DRILL .094 DIA THRU
2 HOLES 180° APART
& LOCATED WITHIN .010
OTP

**SECTION A-A**

DRILL .092-.094 DIA THRU
CSK 90° TO DEPTH SHOWN
24 HOLES EQUALLY SPACED
& LOCATED WITHIN .005
OTP

ENLARGED VIEW
2 X SIZE

AXIS **A-A** ESTABLISHED BY DIA **C**
FEATURES SHALL BE CONCENTRIC & SQUARE
ABOUT AXIS **A-A** WITHIN THE FIR. SPECIFIED BY **B**

ANGULAR LOCATION OF SLOTS AS A UNIT, HOLES IN THEIR
RESPECTIVE PATTERNS AS UNITS, SPLINE, AND GEAR
MAY VARY FREELY

LY THE ITEM DEFINED BY THIS DRAWING AND IDENTIFIED
THE ALLISON PART NUMBER HAS BEEN TESTED AND APPROVED
ALLISON FOR USE IN ALLISON AIRCRAFT ENGINES
SEE PARTS LIST OR PARTS CATALOG FOR SPECIFIC MODELS)
SUBSTITUTE ITEM OR ONE FROM ANOTHER SOURCE SHALL NOT
USED WITHOUT PRIOR TEST AND APPROVAL BY ALLISON
GINEERING SOURCE APPROVAL REQUIRED FOR FORGING,
SO FOR FINISHED PART
TER QUALIFICATION AND APPROVAL BY ALLISON NO CHANGE SHALL
MADE IN THE MANUFACTURING PROCESSES OR MANUFACTURING
RCES WHICH WILL AFFECT THE RELIABILITY OF THIS PART

MACHINE ALL OVER PEEN GEAR WEB AND ADJACENT
FILLETS TO EPS 12120 REMAINING SURFACES MAY BE
PEENED TO EPS 12120 UNLESS SPECIFICALLY CONTROLLED BY A √ SYMBOL

BREAK SHARP EDGES .010 UOS

CASE HARDEN WHERE SHOWN
ROCKWELL HARDNESS CASE C60 MIN (OR EQUIVALENT)
CORE C30 MIN

FORGING SHALL CONFORM TO EIS-502

SURFACE CHARACTERISTICS NOT CONTROLLED BY A √
SYMBOL SHALL BE COMMENSURATE WITH GOOD MANU-
FACTURING PRACTICES WHICH PRODUCE ACCEPTABLE
QUALITY LEVELS.

ber (6829395).

B

# APPENDIX II

## SAMPLE PROCESS ROUTING SHEETS

This appendix consists of sample process routing sheets for a full form ground fillet gear (EX-78772, Figure 129) and for a protuberant hobbed gear (EX-78776, Figure 130). The processing routings for all 16 fatigue test gear part numbers were identical except for the changes required by the two root fillet configurations, as shown in these samples, and for the difference in carburized case depth required by the two diametral pitches.

ROUTE SHEET

| | | | PART OR TOOL NUMBER EX-78772 | REV |
|---|---|---|---|---|
| | | | NEXT ASSEMBLY NUMBER | REV |

| SHEET 1 OF 5 | | PART NAME FATIGUE TEST GEAR | DRAWING NUMBER | REV |
|---|---|---|---|---|
| WRITTEN BY | DATE 9-8-65 | MATERIAL SPEC – SIZE AMS 6265 Forged Bar 5" dia. x 1" long | | |
| APPROVED BY | DATE | MATERIAL SUBSTITUTION | MODEL NO EX-78772 | REV B |

| OPER NO | DEPT | OPERATION | TOOL NO | TOOL CODE | MACHINE |
|---|---|---|---|---|---|
| 1 | 856S | Machine to Sketch Oper. 1 and deburr. | | | Lathe |
| 3 | 819 | Inspect and attach serial number. Start log of PCI 8000 and required information. Forward parts to Dept. 846. | | | |
| 5 | 846 | Hold until all 16 lots of gears are ready to core harden. | | | |
| 7 | 862F | Harden at 1750° and temper per EPS 202 and PCI 8000 for control. C34 - C38. CAUTION: Core harden all 16 lots of gears at same time. | | | Furnace |
| 9 | 856 | Gritblast. | | | |
| 11 | 819 | Rockwell and magnaflux. | | | |
| 13 | 856S | Machine to sketch Oper. #13 and deburr. Transfer tag. | | | Lathe |

Figure 129. Typical Routing Sheet for Full Form Ground Fillet Gear, EX-78772 (Sheet 1 of 9).

ROUTE SHEET

| | | | PART OR TOOL NUMBER EX-78772 | REV |
|---|---|---|---|---|
| | | | NEXT ASSEMBLY NUMBER | REV |

PART NAME: Same as Sheet #1

MATERIAL SPEC – SIZE — DRAWING NUMBER / REV

MATERIAL SUBSTITUTION — MODEL NO / REV

SHEET 2 OF 5

WRITTEN BY — DATE 9-9-65

APPROVED BY — DATE

| OPER NO | DEPT | OPERATION | TOOL NO | TOOL CODE | MACHINE |
|---|---|---|---|---|---|
| 15 | 858 | Grind to sketch oper. #15 and deburr. Transfer tag. | | | Grind |
| 17 | 854 | Rough hob to 4.4367 + .000 - .002" over (2) .288" dia. pins. Hob (5) pieces of this P/N at same time. CAUTION: Use proper hob. Deburr. Transfer tag. | 8-17429 SPT 2603 | Bush Hob | Gear Hob |
| 19 | 854 | Round break gear teeth .031 - .046." | | | |
| 21 | 819 | Gear Lab. Inspect and forward to Dept. 846. Etch S/N on part for heat treat operations. | | | |
| 23 | 846 | Hold until all (16) lots of gears are ready for carburizing. | | | |
| 25 | 862F | Carburize and anneal per EPS 202 - .035 - .045" effective case depth. Use FCI 8000 for control. CAUTION: Carburize (8) lots of gears requiring this case depth at the same time. | | | Furnace |

Figure 129. Typical Routing Sheet for Full Form Ground Fillet Gear, EX-78772 (Sheet 2 of 9).

ROUTE SHEET

| | | PART OR TOOL NUMBER EX-78772 | REV |
|---|---|---|---|
| | | NEXT ASSEMBLY NUMBER | REV |
| | | DRAWING NUMBER | REV |
| | | MODEL NO | REV |

| SHEET 3 OF 5 | PART NAME Same as Sheet #1 |
|---|---|
| WRITTEN BY — DATE 9-9-65 | MATERIAL SPEC – SIZE |
| APPROVED BY — DATE | MATERIAL SUBSTITUTION |

| OPER NO | DEPT | OPERATION | TOOL NO | TOOL CODE | MACHINE |
|---|---|---|---|---|---|
| 27 | 862c | Copper plate all over per PCI 2001. Copper plate (16) lots of gears at the same time. | | | Plating |
| 29 | 862y | Harden and temper per EPS-202 and PCI 8000 for control. CAUTION: All (16) lots of gears to be heat treated at the same time. | | | Furnace |
| 31 | 859 | Gritblast. | | | |
| 33 | 862c | Strip copper plating per PCI 2001. Strip all (16) lots of gears at the same time. | | | Plating |
| 35 | 859 | Mask and shotblast gear teeth only with 80 grit chilled shot. | | | |
| 37 | 819 | Rockwell and magnaflux. | | | |
| 38 | 819 | Gear lab. Inspect gear and record information. | | | |
| 39 | 858 | Grind to sketch Oper. #39 and deburr. Transfer tag. See that S/N is etched back on part after grinding. | 8-17428 | Arbor | Grind |

Figure 129.  Typical Routing Sheet for Full Form Ground Fillet Gear, EX-78772 (Sheet 3 of 9).

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# ROUTE SHEET



| | | | PART OR TOOL NUMBER EX-78772 | REV |
| NEXT ASSEMBLY NUMBER | | | | REV |
| DRAWING NUMBER | | | | REV |
| MODEL NO | | | | REV |

SHEET 4 OF 5  PART NAME: Same as Sheet #1

WRITTEN BY ___ DATE ___  MATERIAL SPEC - SIZE

APPROVED BY ___ DATE ___  MATERIAL SUBSTITUTION

| OPER NO | DEPT | OPERATION | TOOL NO | TOOL CODE | MACHINE |
|---|---|---|---|---|---|
| 41 | 819 | Remove metal tag and etch S/N on web. | | | |
| 43 | 862F | Stress relieve per EPS 202 and PCI 8000. | | | Furnace |
| 45 | 866 | Etch mark teeth per B/P. | | | Sub Assy |
| 47 | 819 | Gear Lab. Inspect pin size, root dia., root radius and record. | | | |
| 49 | 854 | Finish grind gear to 4.3999-4.3979"dia. over (2) .288"dia. pins. Grind (5) pieces of this P/N at the same time. Use 10" grinding wheel TA46G10VB. Down feed .0005 - .001"max. per pass of grinding wheel. Transfer tag. | 8-17428 8-17627 | Arbor Forms | P & W Gear Grind |
| 50 | 819 | Gear lab inspect gear and record information. | | | |
| 51 | 862F | Stress relieve per EPS 202 and PCI 8000. | | | Furnace |
| 53 | 862C | Nital etch per EIS 1510. | | | Plating |
| 54 | 819 | Red line 4 teeth. | | | |

Figure 129. Typical Routing Sheet for Full Form Ground Fillet Gear, EX-78772 (Sheet 4 of 9).

ROUTE SHEET

| | | | | PART OR TOOL NUMBER | | REV |
|---|---|---|---|---|---|---|
| | | | | EX-78772 | | |
| | | PART NAME | | NEXT ASSEMBLY NUMBER | | REV |
| | | Same as Sheet #1 | | | | |
| SHEET 5 OF 5 | | MATERIAL SPEC - SIZE | | DRAWING NUMBER | | REV |
| WRITTEN BY | DATE 9-9-65 | MATERIAL SUBSTITUTION | | MODEL NO | | REV |
| APPROVED BY | DATE | | | | | |

| OPER NO | DEPT | OPERATION | TOOL NO | TOOL CODE | MACHINE |
|---|---|---|---|---|---|
| 55 | 859 | Mark and shot peen gear teeth only per EPS 12140 followed by EPS 12176. Shot peen (16) lots of gears at the same time if possible; if not, shot peen consecutively with case setup. | | | |
| 57 | 854 | Finish hone gear. Use 280 grit hone. | 8-17428 SPT-2606 | Arbor Hone | Gear Hone |
| 59 | 819 | Inspect gear and record information. | | | |
| 61 | 854 | Mount (5) pieces on arbor, matching marked teeth. Remove (12) teeth per B/P. Down feed .0005 - .001" max. per pass of grinding wheel. | 8-17428 | Arbor | Gear Grind |
| 63 | 862P | Stress relieve per EPS 202 and PCI 8000. | | | Furnace |
| 65 | 819 | Magnaflux. | | | |
| 67 | 819 | Inspect for black oxide. | | | |
| 69 | IPO BC5 | Black oxide per AMS-2485. | | | |
| 71 | 819 | Inspect and identify. | | | |

Figure 129. Typical Routing Sheet for Full Form Ground Fillet Gear, EX-78772 (Sheet 5 of 9).

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Figure 129. Typical Routing Sheet for Full Form Ground Fillet Gear, EX-78772 (Sheet 6 of 9).

Figure 129. Typical Routing Sheet for Full Form Ground Fillet Gear, EX-78772 (Sheet 7 of 9).

Figure 129. Typical Routing Sheet for Full Form Ground Fillet Gear, EX-78772 (Sheet 8 of 9).

Figure 129. Typical Routing Sheet for Full Form Ground Fillet Gear, EX-78772 (Sheet 9 of 9).

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ROUTE SHEET

| | | | PART OR TOOL NUMBER | REV |
|---|---|---|---|---|
| | | | EX-78776 | |
| | | | NEXT ASSEMBLY NUMBER | REV |
| SHEET 1 OF 5 | | | DRAWING NUMBER | REV |
| WRITTEN BY | DATE 9-8-65 | | MODEL NO | REV |
| APPROVED BY | DATE | | EX-78776 | B |

PART NAME FATIGUE TEST GEAR
MATERIAL SPEC – SIZE  AMS 6265  Forged Bar  5" dia x 1" long
MATERIAL SUBSTITUTION

| OPER NO | DEPT | OPERATION | TOOL NO | TOOL CODE | MACHINE |
|---|---|---|---|---|---|
| 1 | 8563 | Machine to sketch Oper. 1 and deburr. | | | Lathe |
| 3 | 819 | Inspect and attach serial number. Start log of FCI 8000 and required information. Forward parts to Dept. 846. | | | |
| 5 | 846 | Hold until all 16 lots of gears are ready to core harden. | | | |
| 7 | 862F | Harden at 1750° and temper per EPS 202 and FCI 8000 for control. C34-C38. CAUTION: Core harden all 16 lots of gears at same time. | | | Furnace |
| 9 | 859 | Grit blast. | | | |
| 11 | 819 | Rockwell and magnaflux. | | | |
| 13 | 8563 | Machine to sketch Oper. #13 and deburr. Transfer tag. | | | Lathe |

Figure 130. Typical Routing Sheet for Protuberant Hobbed Gear, EX-78776 (Sheet 1 of 9).

ROUTE SHEET

| | | PART OR TOOL NUMBER EX 78776 | REV |
| | | NEXT ASSEMBLY NUMBER | REV |

PART NAME: Same as Sheet #1

MATERIAL SPEC - SIZE | DRAWING NUMBER | REV

MATERIAL SUBSTITUTION | MODEL NO | REV

SHEET 2 OF 5

WRITTEN BY | DATE 9-9-65
APPROVED BY | DATE

| OPER NO | DEPT | OPERATION | TOOL NO | TOOL CODE | MACHINE |
|---|---|---|---|---|---|
| 15 | 858 | Grind to sketch Oper. #15 and deburr. Transfer tag. | | | Grind |
| 17 | 854 | Rough bob to 4.4367 + .000 - .002" over (2) .288" dia. pins. Bob 5 pcs. of this P/N at same time.<br>CAUTION: Use proper bob.<br>Deburr. Transfer tag. | 8-17429<br>SPT-2604 | Bush<br>Bob | Gear<br>Bob |
| 19 | 854 | Round break gear teeth .031 - .046". | | | |
| 21 | 819 | Gear Lab.<br>Inspect and forward to Dept. 846.<br>Etch serial number for heat treat operations. | | | |
| 23 | 846 | Hold until all 16 lots of gears are ready for carburizing. | | | |
| 25 | 862P | Carburize and anneal per EPS 202  .035-.045" effective case depth.<br>Use FCI 8000 for control.<br>CAUTION: Carburize 8 lots of gears requiring this case depth at the same time. | | | Furnace |

Figure 130. Typical Routing Sheet for Protuberant Hobbed Gear, EX-78776 (Sheet 2 of 9).

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ROUTE SHEET

| SHEET 3 OF 5 | PART NAME | Same as Sheet #1 |
| --- | --- | --- |
| WRITTEN BY | MATERIAL SPEC - SIZE | |
| DATE 9-9-65 | | |
| APPROVED BY | MATERIAL SUBSTITUTION | |
| DATE | | |

| OPER NO | DEPT | OPERATION | TOOL NO | TOOL CODE | MACHINE |
| --- | --- | --- | --- | --- | --- |
| 27 | 862C | Copper plate all over per PCI 2001.<br>Copper plate 16 lots of gears at the same time. | | | Plating |
| 29 | 862F | Harden and temper per EPS 202 and PCI 8000 for control.<br>CAUTION: All 16 lots of gears to be heat treated at the same time. | | | Furnace |
| 31 | 859 | Gritblast. | | | |
| 33 | 862C | Strip copper plating per PCI 2001.<br>Strip all 16 lots of gears at the same time. | | | Plating |
| 35 | 859 | Shotblast gear teeth with 80 grit chilled shot. | | | |
| 37 | 819 | Rockwell and magnaflux. | | | |
| 38 | 819 | Gear Lab.<br>Inspect gear and record information. | | | |

Figure 130. Typical Routing Sheet for Protuberant Hobbed Gear, EX-78776 (Sheet 3 of 9).

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ROUTE SHEET

| SHEET | 4 | OF | 5 | PART NAME | Same as sheet #1 | | PART OR TOOL NUMBER EX-78776 | REV |
|---|---|---|---|---|---|---|---|---|
| | | | | | | NEXT ASSEMBLY NUMBER | REV |
| WRITTEN BY | | DATE 9-9-65 | | MATERIAL SPEC - SIZE | | DRAWING NUMBER | REV |
| APPROVED BY | | DATE | | MATERIAL SUBSTITUTION | | MODEL NO | REV |

| OPER NO | DEPT | OPERATION | TOOL NO | TOOL CODE | MACHINE |
|---|---|---|---|---|---|
| 39 | 862C | Mask and solution machine gear teeth only per EPS 13066, .002" max. stock removal. Solution machine 8 lots of gears requiring this operation at the same time, or as near alike as possible, removing the same amount of stock on all lots. | | | |
| 40 | 819 | Gear Lab. Inspect gear and record information. | | | |
| 41 | 859 | Mask and shot peen gear teeth only per EPS-12140 followed by EPS 12176. Shot peen lo lots of gears at the same time if possible; if not, shot peen consecutively with one setup. | | | |
| 43 | 858 | Grind per sketch Oper. #43 and deburr. Transfer tag. Etch 3/N on part after grinding. | 8-17428 | Arbor | Grind |
| 45 | 819 | Remove metal tag and etch S/N on web. | | | |
| 47 | 862F | Stress relieve per EPS 202 and PCI 8000. | | | Furnace |
| 49 | 866 | Etch mark gear teeth per B/P. | | | Sub Assy |

Figure 130. Typical Routing Sheet for Protuberant Hobbed Gear, EX-78776 (Sheet 4 of 9).

ROUTE SHEET

| | | PART NAME | Same as Sheet #1 | PART OR TOOL NUMBER | | REV |
|---|---|---|---|---|---|---|
| SHEET 5 OF 5 | | | | EX 78776 | | |
| WRITTEN BY | DATE 9-9-65 | MATERIAL SPEC – SIZE | | NEXT ASSEMBLY NUMBER | | REV |
| APPROVED BY | DATE | MATERIAL SUBSTITUTION | | DRAWING NUMBER | | REV |
| | | | | MODEL NO | | REV |

| OPER NO | DEPT | OPERATION | TOOL NO | TOOL CODE | MACHINE |
|---|---|---|---|---|---|
| 51 | 854 | Finish grind gear. Grind 5 pieces of this P/N at the same time. Deburr." Use 8.66" dia. wheel 38A605KVBE. Down feed .0005 - .001" max. per pass of grinding wheel. | 8-17428 <br> 8-17459 | Arbor <br> Pitch <br> Block | Mag Gear Grind |
| 52 | 819 | Gear Lab. Inspect gear and record information and red line 4 teeth. | | | |
| 53 | 862F | Stress relieve per EPS 202 and PCI 8000. | | | Furnace |
| 54 | 862C | Nital etch per EIS 1510 | | | Plating |
| 55 | 854 | Mount 5 pieces on arbor matching marked teeth. Remove 12 teeth per B/P. Down feed .0005-.001" max. | 8-17428 | Arbor | Gear Grind |
| 57 | 862F | Stress relieve per EPS 202 and PCI 8000. | | | Furnace |
| 59 | 819 | Magnaflux | | | |
| 61 | 819 | Inspect for black oxide. | | | |
| 63 | IPO BC5 | Black oxide per AMS 2485. | | | |
| 65 | 819 | Inspect and identify. | | | |

Figure 130. Typical Routing Sheet for Protuberant Hobbed Gear, EX-78776 (Sheet 5 of 9).

4.583
Diameter

1.250
Diameter

0.750

Dimensions in Inches
Drawing To Scale

ALL THREE PLACE DECIMALS ARE ± .010 UNLESS OTHERWISE SPECIFIED
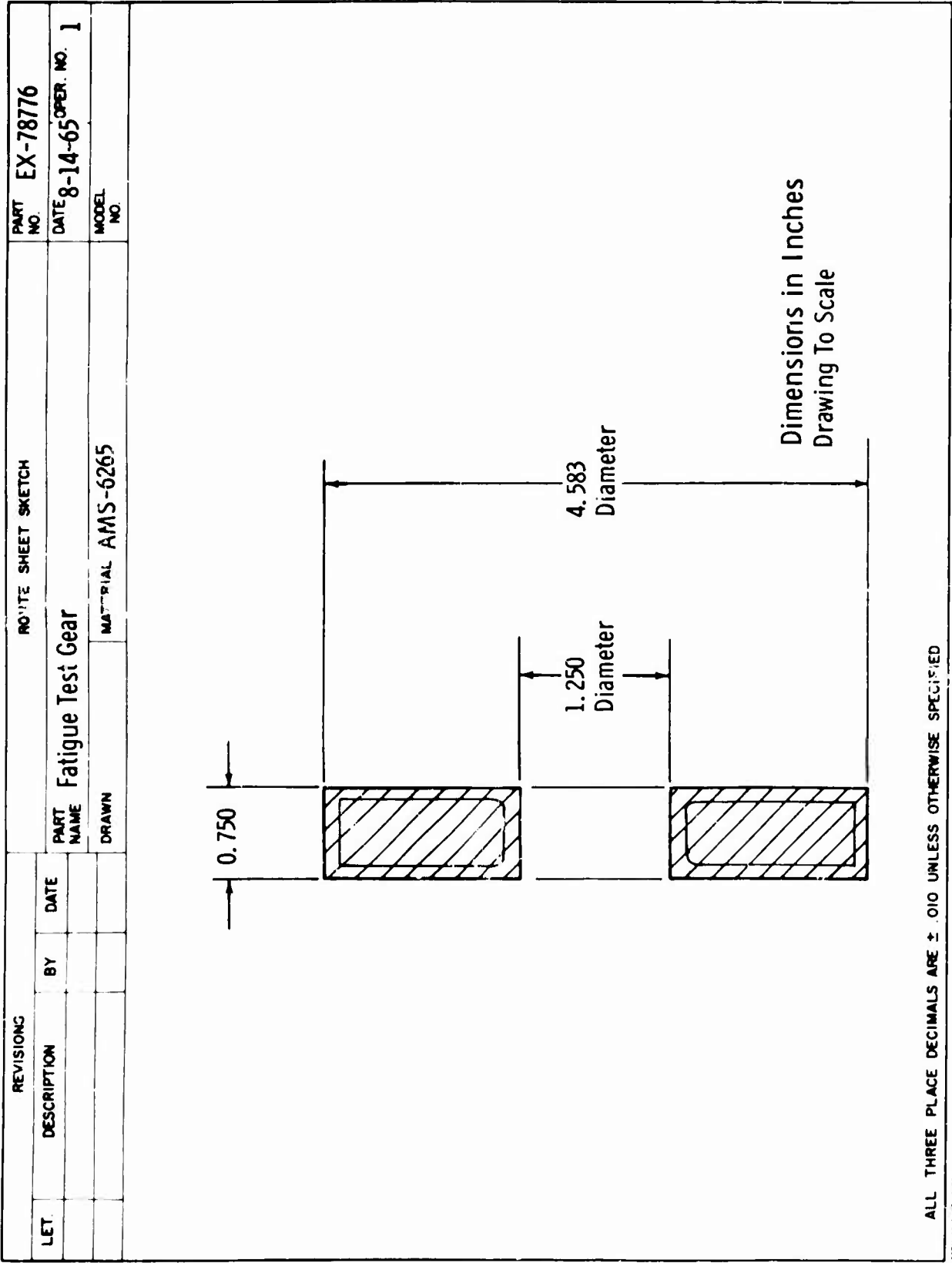
Figure 130.   Typical Routing Sheet for Protuberant Hobbed Gear,  EX-78776 (Sheet 6 of 9).

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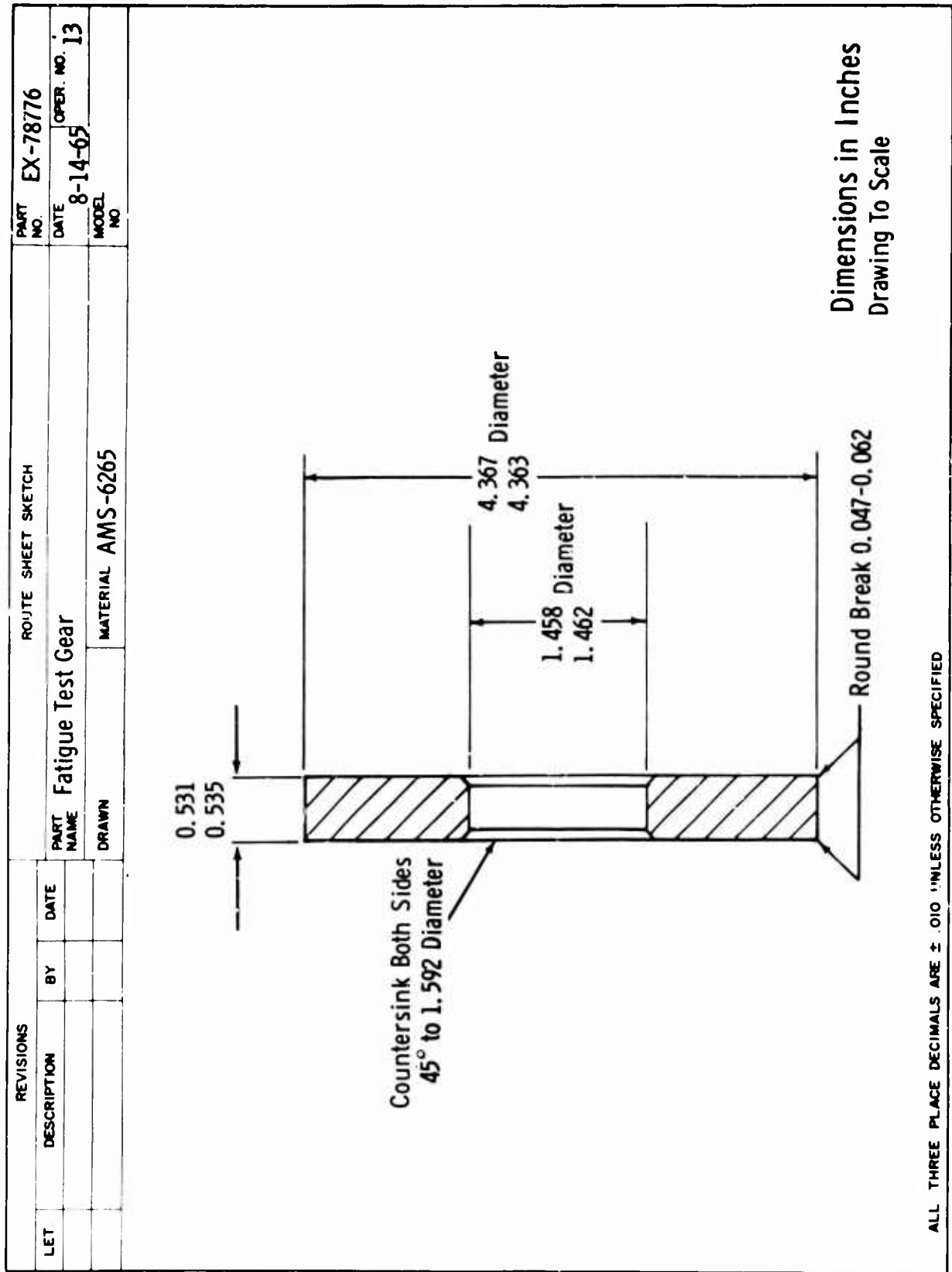
Figure 130. Typical Routing Sheet for Protuberant Hobbed Gear, EX-78776 (Sheet 7 of 9).
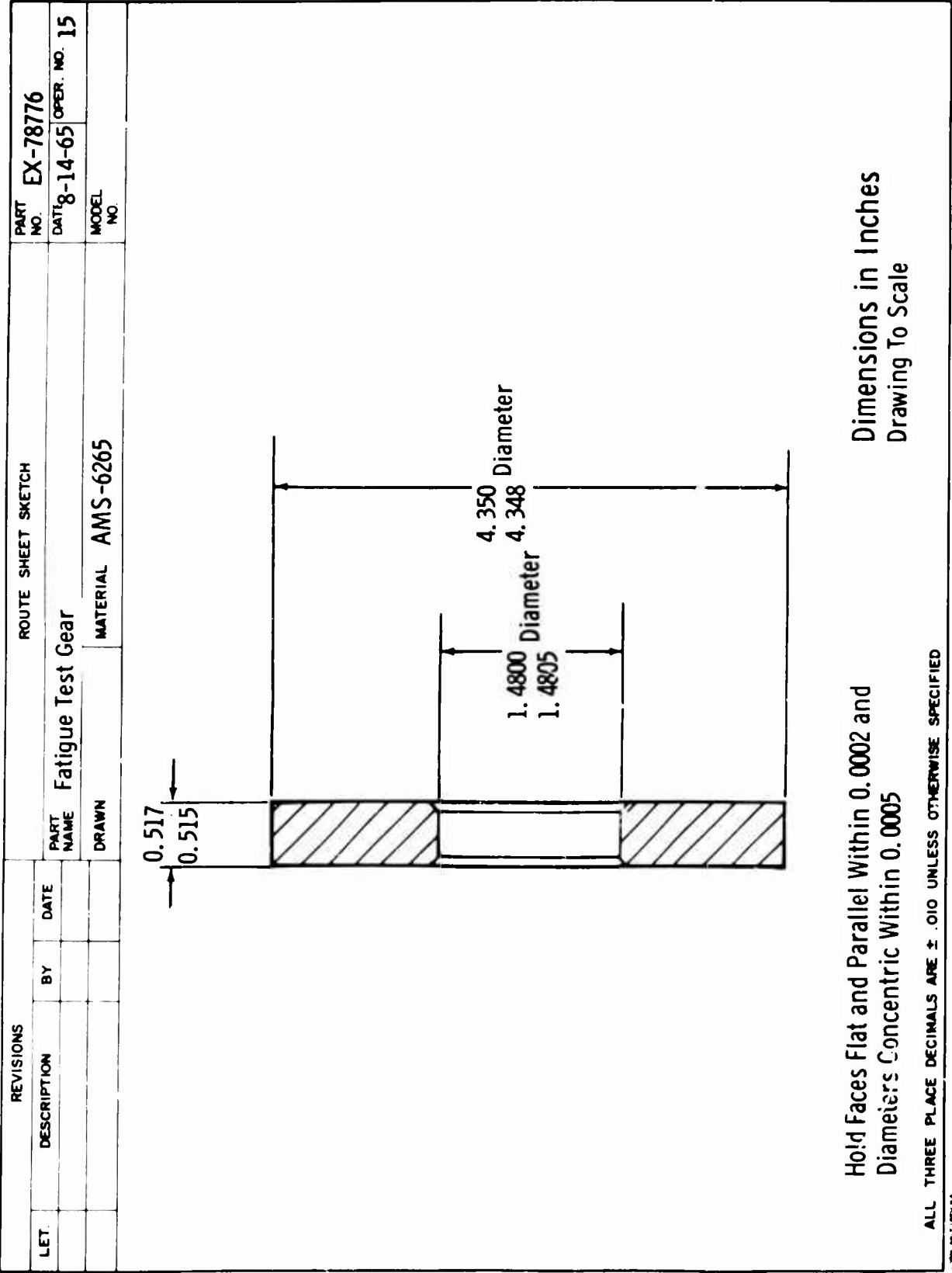
Figure 130. Typical Routing Sheet for Protuberant Hobbed Gear, EX-78776 (Sheet 8 of 9).
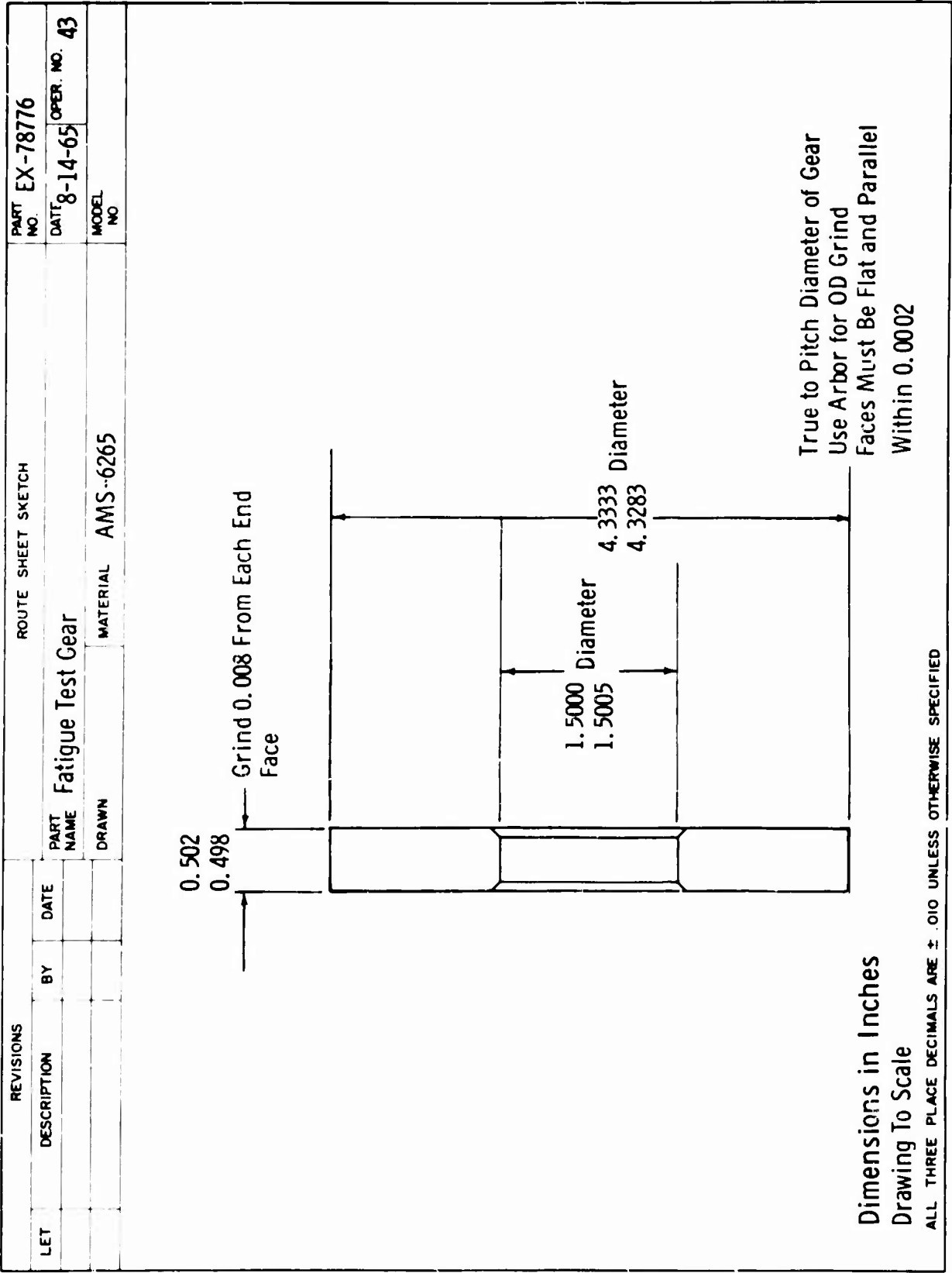
Figure 130. Typical Routing Sheet for Protuberant Hobbed Gear, EX-78776 (Sheet 9 of 9).

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## MATHEMATICAL DESCRIPTION OF STATISTICAL TREATMENT OF TEST DATA

This appendix consists of a detailed description of the mathematical model developed to linearize the test data, its substantiation, its use to determine an endurance limit, and the determination of the variability associated with this endurance limit. A description of the method used to determine the significance of main effects and interactions for the four designed experiment variables is included. Finally, a mathematical equation developed to assign numerical values to the four geometric factors studied is described.

### DERIVATION OF S/N CURVE

#### Analytical Model

There were two requisites for the mathematical model; it should linearize the relationship between cycles-to-failure and stress to define the endurance limit accurately, and it should make the variance of the transformed cycles equal within the range of interest for stress to make tests of significance meaningful.

The mathematical model developed is:

$$\text{Life}_T = \left(\frac{1}{K}\right)^C = A + Bx$$

where

K = kilocycles to failure
x = applied stress
C = linearizing parameter
A and B = constants to be determined by the least squares fitting method

The model was checked against two relatively large sets of data. The transformed data are plotted in Figures 131 and 132. The points and the fitted curve are presented in conventional S/N format in Figures 133 and 134. The linearity of the transformed data is evident by inspection. The homogeneity of variances was checked using Bartlett's test. The stress (or strength) at infinite life is clearly shown at $\text{Life}_T = \left(\frac{1}{K}\right)^C = 0$.

The value of C was selected by trial and error because of time limitations. Further development work is suggested to automate the optimization of C and to investigate an alternate transformation, $\text{Life}_T = \dfrac{1}{\log{(CK)}}$

#### Treatment of Runouts

Runout data were used in one of two ways. If only runouts occurred at any one stress level, the runouts were treated as failures at $10^9$ cycles. Where both runouts and failures occurred at a stress level for any configuration, the data were plotted on normal probability paper using mean ranks to plot the cumulative probability. The points were fitted with a straight line with a slope that best fit all sets of data. The cycles at 50-percent failure represented the average life for all teeth tested at that stress level for the configuration. This value of life, weighted for the associated number of failed teeth, was used in the least squares fit of the complete S/N line.
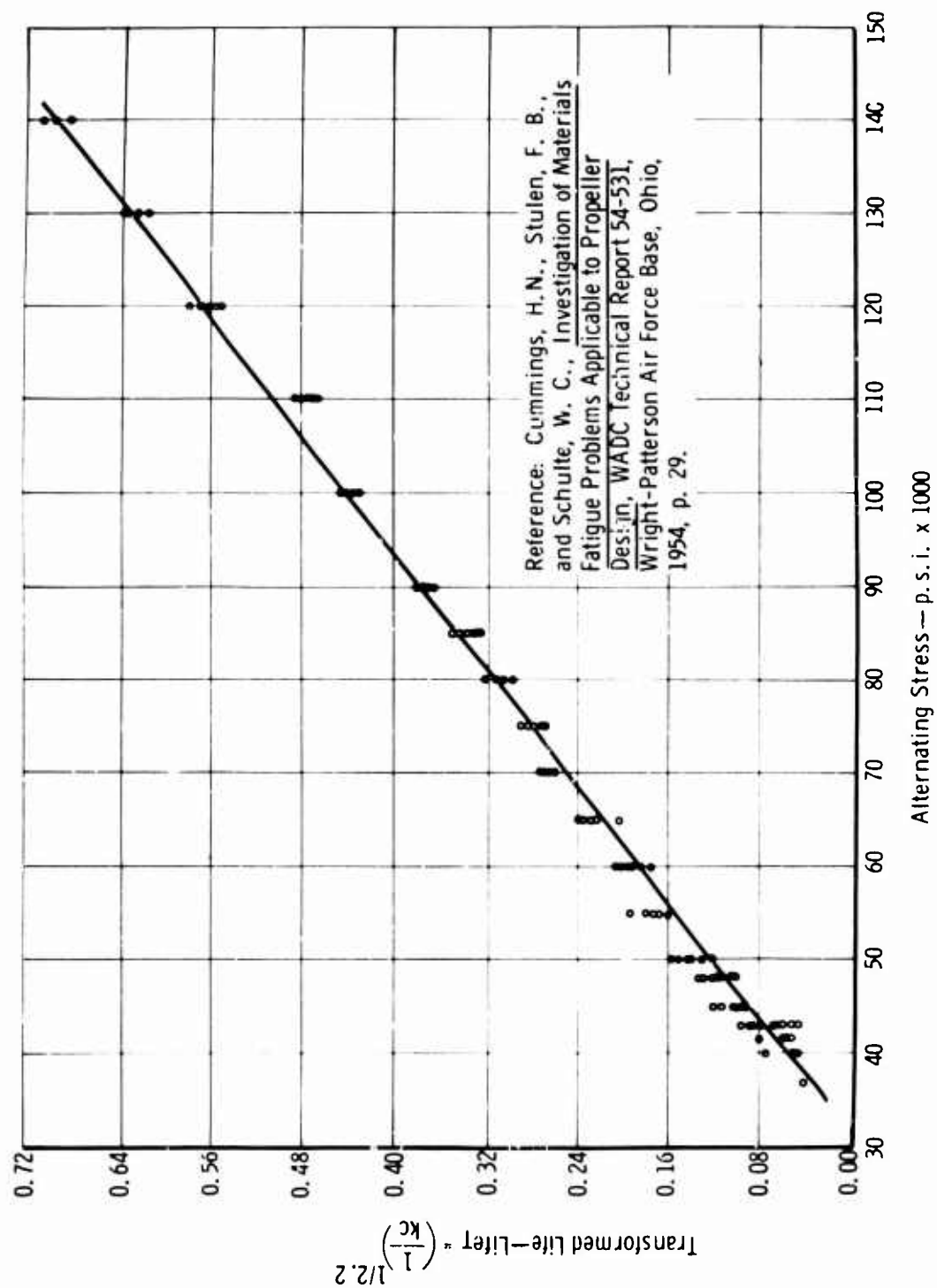
Figure 131. Results of R. R. Moore Tests on Notched 4340 Steel.
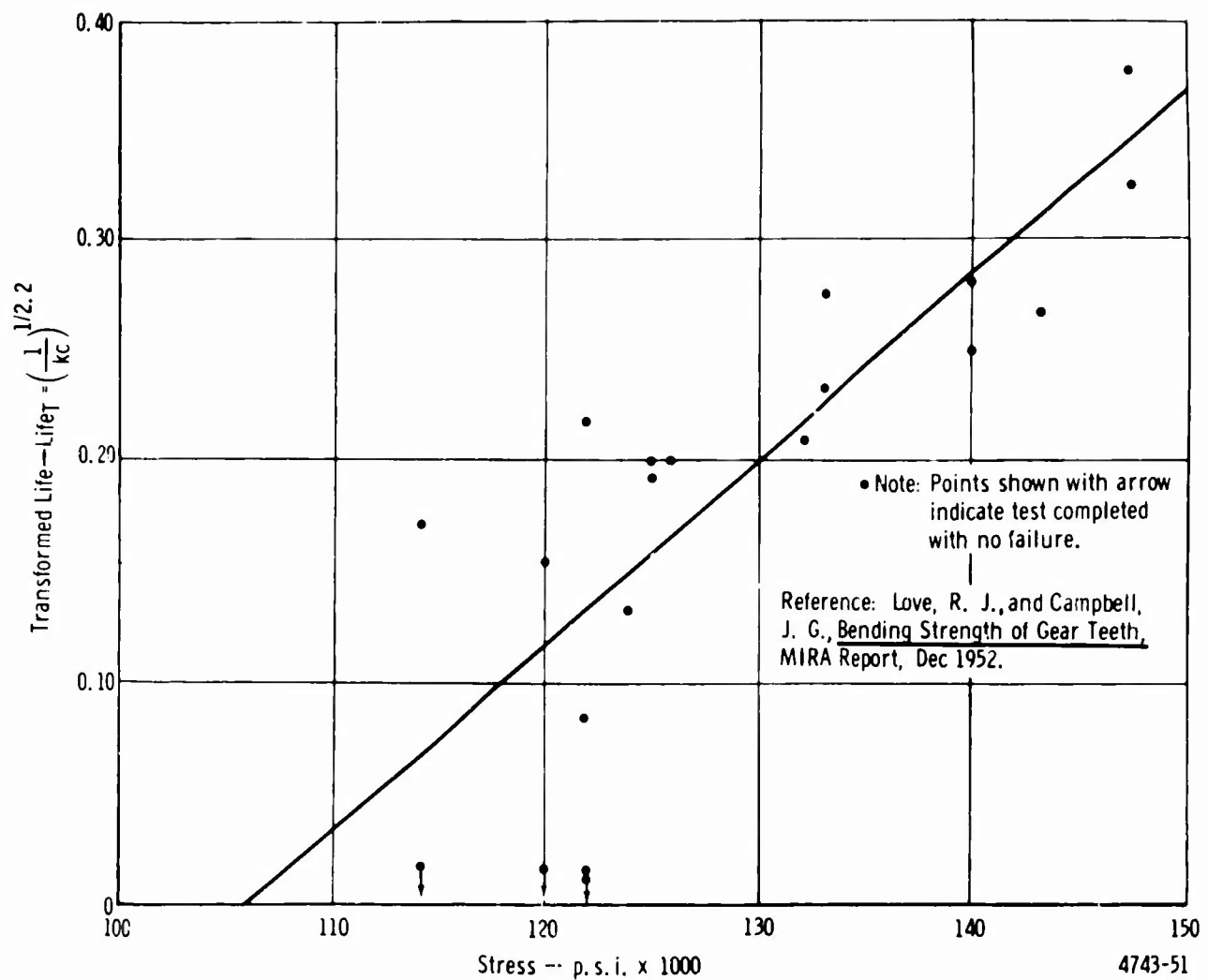
**Figure 132. Transformed Gear Tooth Fatigue Data—British Steel EN 39A.**

Axis labels and annotations within the figure:

Transformed Life—Life$_T$ = $\left(\frac{1}{kc}\right)^{1/2.2}$

Stress -- p.s.i. x 1000

• Note: Points shown with arrow indicate test completed with no failure.

Reference: Love, R. J., and Campbell, J. G., Bending Strength of Gear Teeth, MIRA Report, Dec 1952.

4743-51

## Analysis

The least squares fit of the S/N line for each combination of gear factors represents a solution to the equation Life$_T$ = A + **Bx**. Recalling that the endurance limit occurs at Life$_T$ = 0, it follows that A + Bx = 0 at this point. Subtracting A from both sides of the equation and dividing through by B, and since A is negative, the value of x at the endurance limit is simply A/B.

Each endurance limit A/B has a measure of variability associated with it. This variability is indicated by the scatter in test points about the line, which results from inherent variability in material, processing, and testing factors. The variability or variance, $(\sigma_{A/B})^2$, of each intercept was derived through error propagation techniques (reference 20):

$$(\sigma_{A/B})^2 = \frac{1}{B^2}\sigma_A^2 + \frac{A^2}{B^4}\sigma_B^2 + \frac{2A}{B^3}\sigma_{AB}^2$$

where the components $\sigma_A^2$, $\sigma_B^2$, and $\sigma_{AB}^2$ represent the variance of A, variance of B, and covariance of A and B, respectively. The variances of A, B, and the covariance

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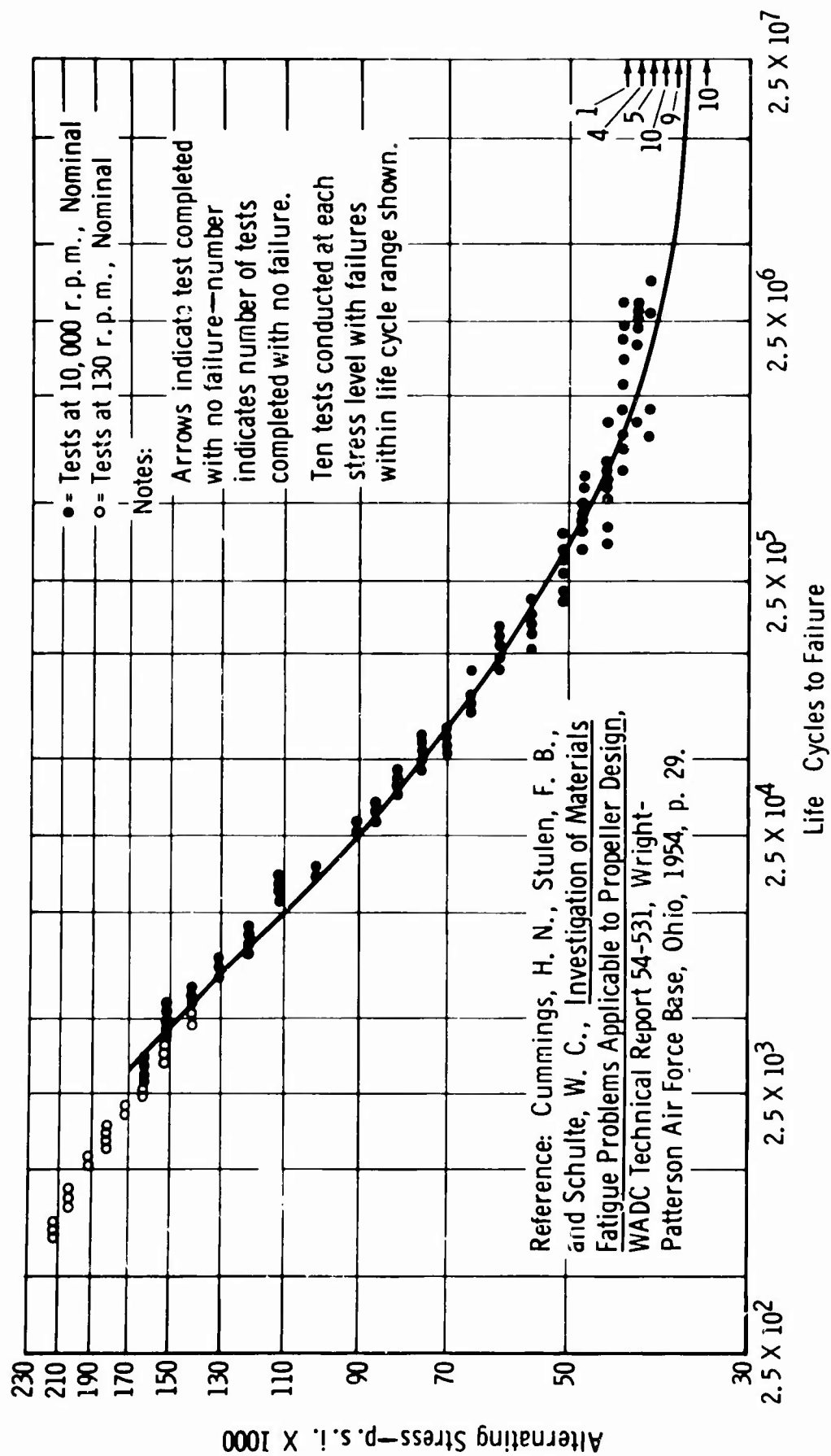
Figure 133. R. R. Moore Rotating Bending Test Data.

= Tests at 10,000 r.p.m., Nominal
= Tests at 130 r.p.m., Nominal

Notes:

Arrows indicate test completed
with no failure—number
indicates number of tests
completed with no failure.

Ten tests conducted at each
stress level with failures
within life cycle range shown.

Reference: Cummings, H. N., Stulen, F. B.,
and Schulte, W. C., Investigation of Materials
Fatigue Problems Applicable to Propeller Design,
WADC Technical Report 54-531, Wright-
Patterson Air Force Base, Ohio, 1954, p. 29.

Life Cycles to Failure

Alternating Stress—p.s.i. X 1000

Figure 134. Gear Tooth Fatigue Data—British Steel EN 39A.

of A and B were evaluated using the techniques presented in reference 3. Briefly, a matrix arising from the least squares solution of A and B is set up and inverted. The inverse elements of the least squares matrix, when multiplied by the variance, $S_e^2$

$$= \frac{\Sigma(\text{Life}_T - A - Bx)^2}{n-2}$$ (where n is the number of test points defining the line) associated with regression, are the variances of A, B, and the covariance of A and B.

To test the significance of main effects and interactions, linear combinations of the 16 endurance limits were computed and then divided by the appropriate standard deviation. The linear combination divided by the standard deviation constitutes the criterion for "t" tests of significance.

STATISTICAL TESTS OF SIGNIFICANCE

The concept of statistical tests of significance arises because of the inherent variability associated with any type of testing. In particular, the variability associated with fatigue testing is large.

If repeat fatigue tests are made under identical test conditions, the computed endurance limits will not be identical, but will be distributed about the average of the computed values. If one or more test conditions are changed (i.e., geometric factors), a criterion may be set up to determine if the magnitude of the change in endurance limits is larger than can be expected due to chance alone—at a preselected probability level.

The criterion established was the "t" test, where "t" is the observed difference in endurance limits generated from two different test conditions. These test conditions were then divided by the standard deviation of the difference:

$$"t" = \frac{EL_1 - EL_2}{\sqrt{S_1^2 + S_2^2}}$$

where

$EL_1$      = the endurance limit associated with the first test condition

$EL_2$      = the endurance limit associated with the second test condition

$S_1^2$ and $S_2^2$ = the variances associated with the respective endurance limits

The critical "t" value is a number based on degrees of freedom (related to number of data points), and some preselected significance level $\alpha$ (an arbitrary risk of making a wrong conclusion). The degrees of freedom for the gear test program was approximately 50. The significance level was selected as $\alpha = 0.05$. Therefore, if the computed "t" was equal to or greater than 2.0, it was concluded that the factor evaluated caused a real (or significant) change in endurance limit. For the mathematical sense, $\alpha$ is defined as the probability that a "t" value larger than the critical "t" will result if the evaluated geometric factor has no true effect on endurance limit; therefore, if a "t" larger than the critical "t" is computed, the odds are 19 to 1 that the effect is real.

Some modification of the "t" tests of significance was necessary because of unequal sample sizes in the 16 combinations of the four geometric factors. The resulting "t" tests are set up by first obtaining the difference between weighted average associated with low and high values assigned to the geometric factors, and then dividing by an approximate standard deviation.

$$"t" = \frac{\left( \dfrac{\Sigma W_L \, EL_L}{\Sigma W_L} - \dfrac{\Sigma W_H \, EL_H}{\Sigma W_H} \right)}{\sqrt{\dfrac{1}{64} \displaystyle\sum_{i=1}^{16} \sigma_i^2}}$$

where

     W   = sample size
     EL = endurance limit
     L    = low
     H   = high

The undefined indices of summation include run numbers to which low values and high values, respectively, have been assigned for the evaluation of any factor or interaction.

Confidence intervals are also based on the same critical "t" values and variances used in tests of significance. Confidence intervals are set up by the equations:

$$LL = EL - "t"_{\alpha/2} \times S_{EL}$$

$$UL = EL + "t"_{\alpha/2} \times S_{EL}$$

For mathematical terms, the probability is $(1-\alpha)$ that the resulting interval will contain the true endurance limit.

An example of a test of significance is provided for the main effect—diametral pitch. For convenience, the following notation is defined:

|   |   |   | High | Low |
|---|---|---|------|-----|
| a | = | diametral pitch | 12 | 6 |
| b | = | pressure angle, degrees | 25 | 20 |
| c | = | root radius | Large | Small |
| d | = | fillet configuration | Full form | Protuberance |

By convention, the presence of a letter (associated with a geometric factor) indicates that the high value is assigned to that factor. The absence of a letter indicates that the low level is assigned to that factor. Further, (1) means that the low level is assigned to all factors. Thus, the configuration ab means gears of 12 diametral pitch, 25-degree pressure angle, small radius, and protuberance ground.

To test the significance of diametral pitch using the notation developed, a linear combination of 16 computed endurance limits was set up.

$$L = 1/8 \left[ a + ab + ac + abc + ad + abd + acd + abcd \right] -$$

$$1/8 \left[ (1) + b + c + bc + d + bd + cd + bcd \right]$$

The first group contains all gear configurations of 12 diametral pitch, and the second group contains all configurations of 6 diametral pitch.

The variance of L, which is the same for all tests, is:

$$L^2 = \frac{1}{64} \left[ \sigma_a^2 + \sigma_{ab}^2 + \ldots + \sigma_{abcd}^2 \right] + \frac{1}{64} \left[ \sigma_{(1)}^2 + \ldots + \sigma_{bcd}^2 \right]$$

A "t" test of significance is set up by dividing L by the standard deviation of $\sigma_L$ or "t" $= \frac{L}{\sigma_L}$.

The four main effects, all two-factor interactions and all three-factor interactions, were tested using this method. The exact linear combination for any specified effect or interaction is found in reference 14 or 29.

## PREDICTIVE EQUATION BASED ON TEST RESULTS

A second objective in the analysis of gear tooth fatigue failures was to develop a single predictive equation incorporating numerical values assigned to the geometric factors in addition to the basic applied load. The technique is as follows:

1. Define a linear mathematical model

$$\text{Life}_T = A + Bx$$

where

$$\text{Life}_T = (1/K)^{1/2.2}$$

K = kilocycles to failure

x = unit stress

2. Redefine the geometric factors

| Factor | Range |
|---|---|
| U1 = pressure angle, degrees | 20 - 25 |
| U2 = diametral pitch | 6 - 12 |
| U3 = dedendum | 1.20 - 1.40 |
| $U4 = \dfrac{\text{minimum fillet radius}}{\text{maximum fillet radius}}$, inch | 0.49 - 0.80 |

The coefficients A and B in the linear model are defined by the geometric factors as follows:

A = (a0 + a1 U1 +... + a4 U4 + a5 U1 U2 +... + a10 U3 U4 + a11 U1 U2 U3 +... + a14 U2 U3 U4)

B = (b0 + b1 U1 +... + b14 U2 U3 U4)

In terms of the refined coefficients, the expanded model is:

$$\text{Life}_T = (1/K)^{1/2.2} = (a0 + a1\ U1 + ... + a14\ U2\ U3\ U4) +$$
$$(b0 + b1\ U1 + ... + b14\ U2\ U3\ U4)\ X$$

The individual coefficients were evaluated using the least squares technique.

The following geometric factors affect fatigue life and are listed in order of decreasing importance:

1. (Pressure angle $\times$ diametral pitch $\times$ dedendum) $\times$ load
2. Pressure angle $\times$ diametral pitch $\times$ dedendum
3. Pressure angle $\times$ diametral pitch
4. Pressure angle $\times$ dedendum $\times$ fillet radius
5. Pressure angle $\times$ fillet radius
6. Pressure angle $\times$ load
7. Pressure angle
8. Dedendum
9. Diametral pitch $\times$ dedendum
10. (Pressure angle $\times$ diametral pitch) $\times$ load
11. Dedendum $\times$ fillet radius

In terms of coding, the finalized equation is:

$$\text{Life}_T = (1/K)^{1/2.2} = 2.27864 - 5.47376 \times 10^{-2}\ (U1) - 1.18640\ (U3) -$$

$$8.97196 \times 10^{-3}\ (U1\ U2) + 1.20233 \times 10^{-1}\ (U1\ U4) -$$

$$3.67334 \times 10^{-2} \ (U2 \ U3) + 4.43879 \times 10^{-1} \ (U3 \ U4) +$$

$$9.11496 \times 10^{-3} \ (U1 \ U2 \ U3) - 1.17884 \times 10^{-1} \ (U1 \ U3 \ U4) \ (load) +$$

$$X \ \{-3.58085 \times 10^{-6} \ (U1) - 1.09015 \times 10^{-6} \ (U1 \ U2) +$$

$$1.75948 \times 10^{-6} \ (U1 \ U2 \ U3)\}$$

The standard deviation $(\sigma_Y)$ associated with the predictive equation is 0.0656.

The equation can be used to predict transformed kilocycles only within the range of interest for applied load values and only within the range of values assigned to the geometric factors from which the equation was derived.

The most efficient use of the predictive equation can be obtained by first computing transformed kilocycles using observed values for the geometric factors and the applied load, and then converting to cycles or kilocycles, as desired. To obtain an approximate confidence interval for kilocycles to failure, add and subtract the quantity $(Z_{a/2} \times 0.0656)$ to and from the calculated Y = transformed kilocycles ($Z_{a/2}$ is a confidence factor to be obtained from a table of areas for the normal distribution). These computed upper and lower limits are then transformed to kilocycles using the same procedures used to convert Y to kilocycles.

The equation, although derived from valid test data, is yet untried in the predictive sense. It may be that additional testing, at more than two levels per geometric factor, may be required to derive a mathematical model suitable for general usage in predicting gear failures.

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## APPENDIX IV

## AGMA CALCULATED STRESS VERSUS LIFE AND TRANSFORMED LIFE

This appendix consists of life versus AGMA calculated stress plots of the fatigue test data points for each of the 16 gear configurations. See Figures 135 through 150. The calculated mean S/N curve fitting the data points is drawn on each plot. Also included are transformed life versus AGMA calculated stress plots of the fatigue test data points for each of the 16 gear configurations. See Figures 151 through 166. Life and transformed life versus alternating stress (R. R. Moore) data are shown in Figures 167 and 168, respectively.

Life Cycles
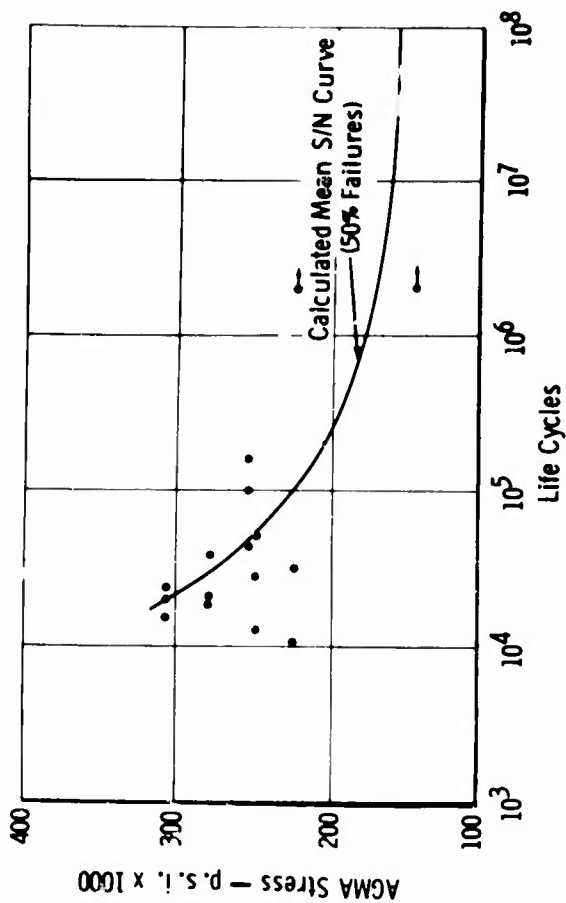
Figure 136. Fatigue Test Results—AGMA Stress Versus Life (EX-78773).

Calculated Mean S/N Curve (50% Failures)

AGMA Stress — p.s.i. x 1000

400  300  200  100

$10^3$  $10^4$  $10^5$  $10^6$  $10^7$  $10^8$

Life Cycles
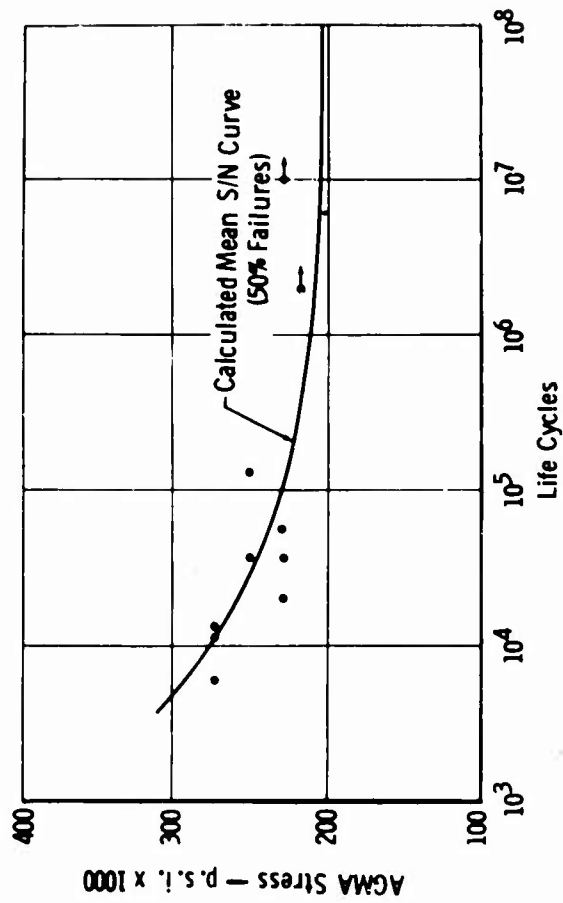
Figure 138. Fatigue Test Results—AGMA Stress Versus Life (EX-78775).

Calculated Mean S/N Curve (50% Failures)

AGMA Stress — p.s.i. x 1000

400  300  200  100

$10^3$  $10^4$  $10^5$  $10^6$  $10^7$  $10^8$

Life Cycles
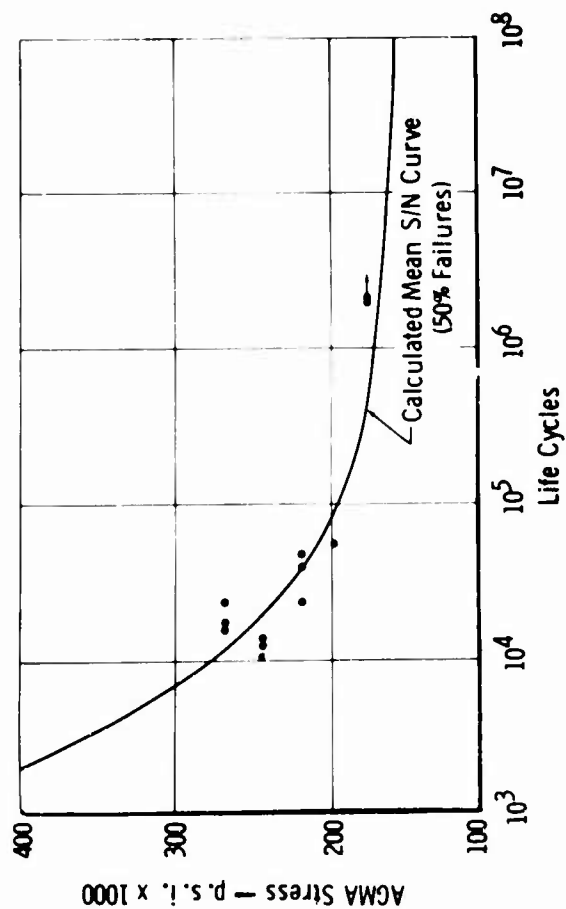
Figure 135. Fatigue Test Results—AGMA Stress Versus Life (EX-78772).

Calculated Mean S/N Curve (50% Failures)

AGMA Stress — p.s.i. x 1000

400  300  200  100

$10^3$  $10^4$  $10^5$  $10^6$  $10^7$  $10^8$

Life Cycles
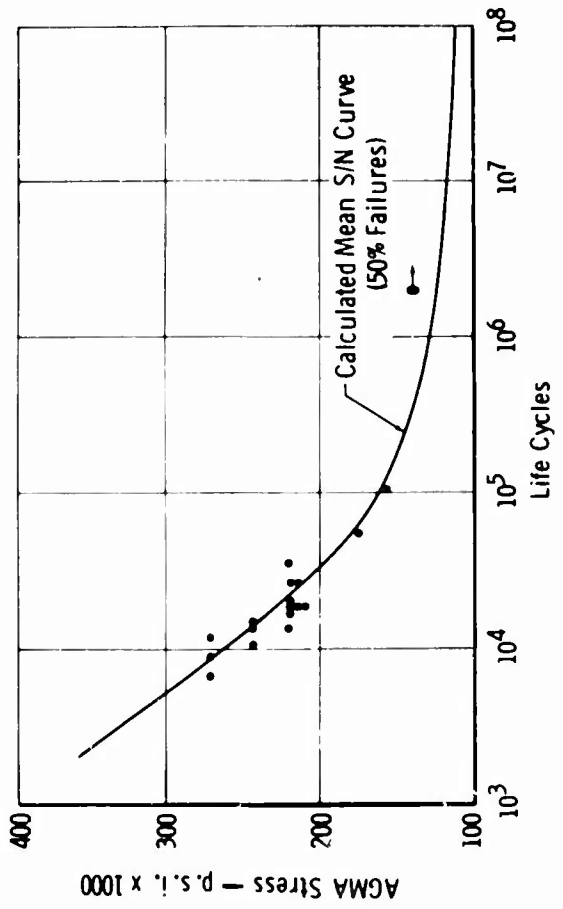
Figure 137. Fatigue Test Results—AGMA Stress Versus Life (EX-78774).

Calculated Mean S/N Curve (50% Failures)

AGMA Stress — p.s.i. x 1000

400  300  200  100

$10^3$  $10^4$  $10^5$  $10^6$  $10^7$  $10^8$

216
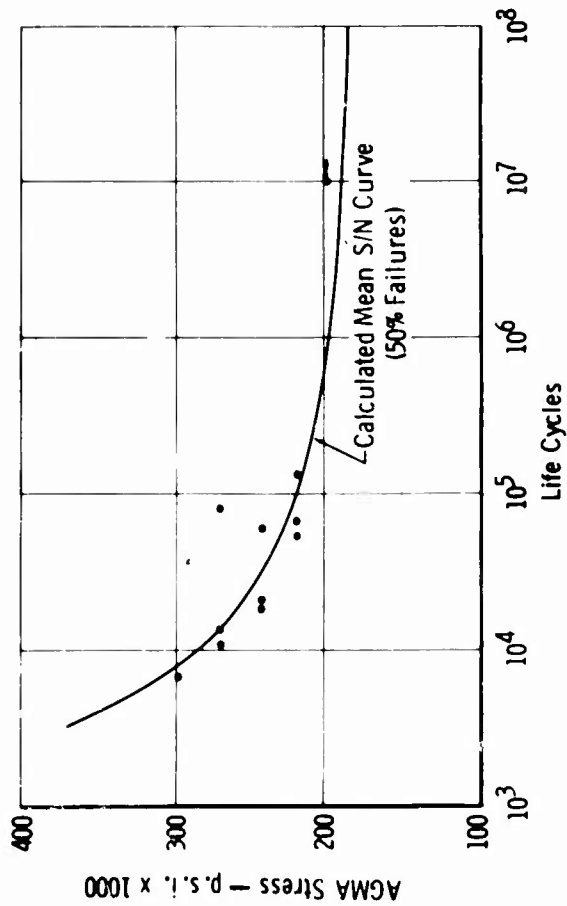
Figure 139. Fatigue Test Results—AGMA Stress
Versus Life (EX-78776).

Figure 140. Fatigue Test Results—AGMA Stress
Versus Life (EX-78777).
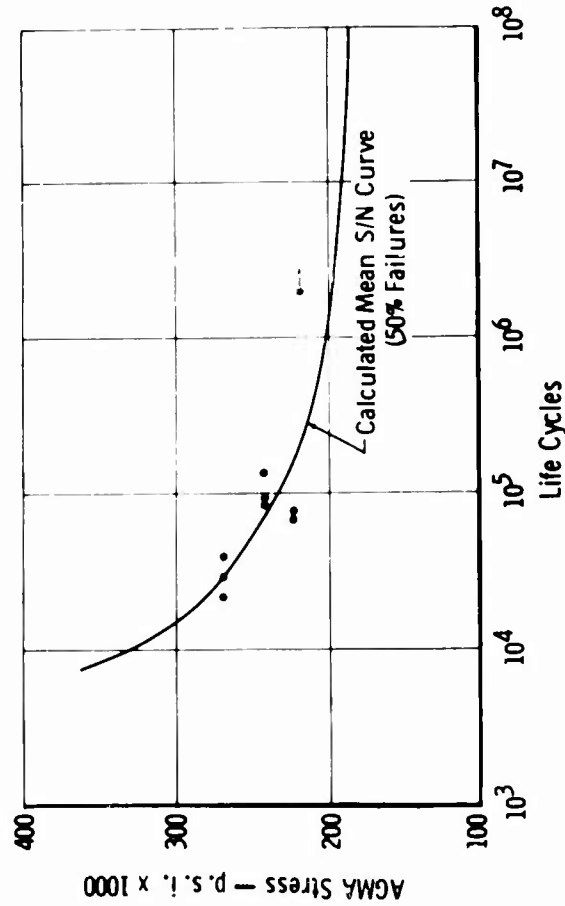
Figure 141. Fatigue Test Results—AGMA Stress
Versus Life (EX-78778).
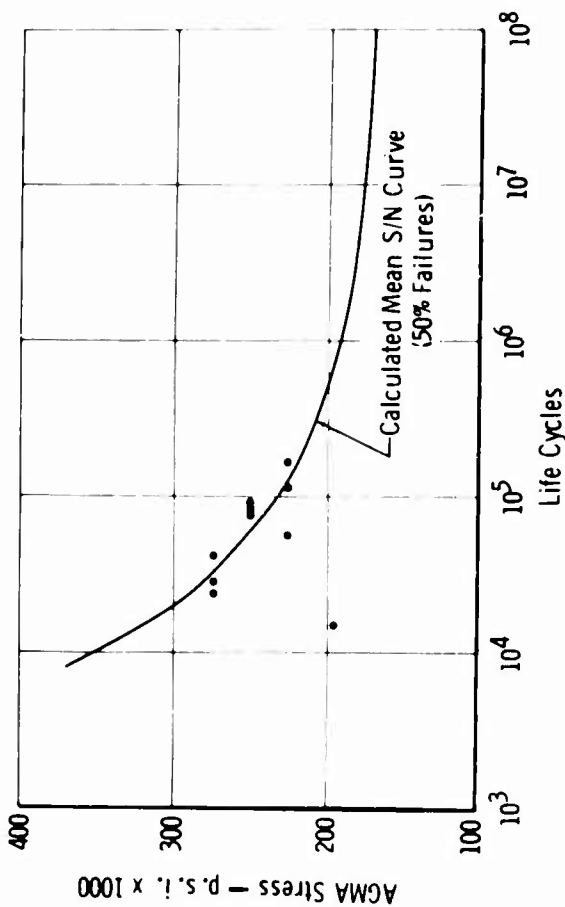
Figure 142. Fatigue Test Results—AGMA Stress
Versus Life (EX-78779).

217
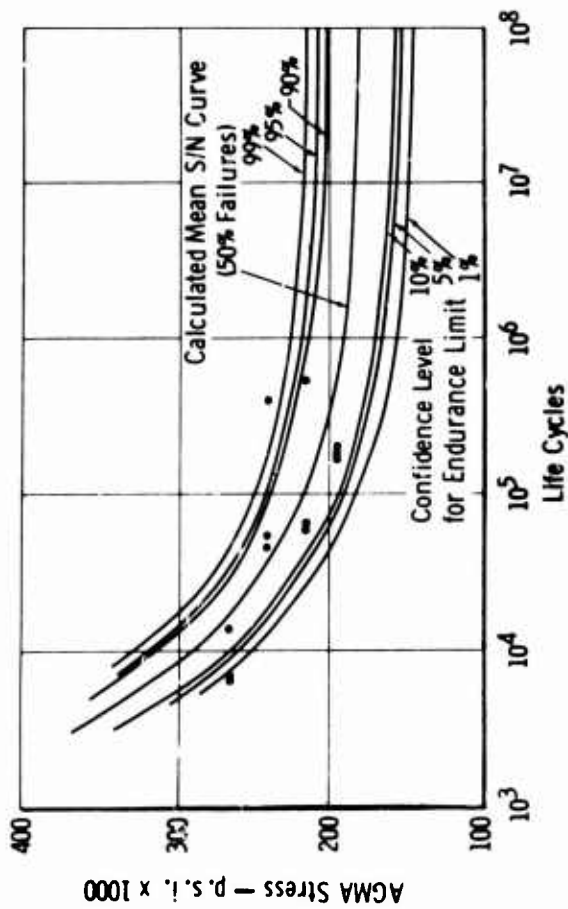
Figure 143. Fatigue Test Results - AGMA Stress Versus Life (EX-78780).



Figure 144. Fatigue Test Results—AGMA Stress Versus Life (EX-78781).
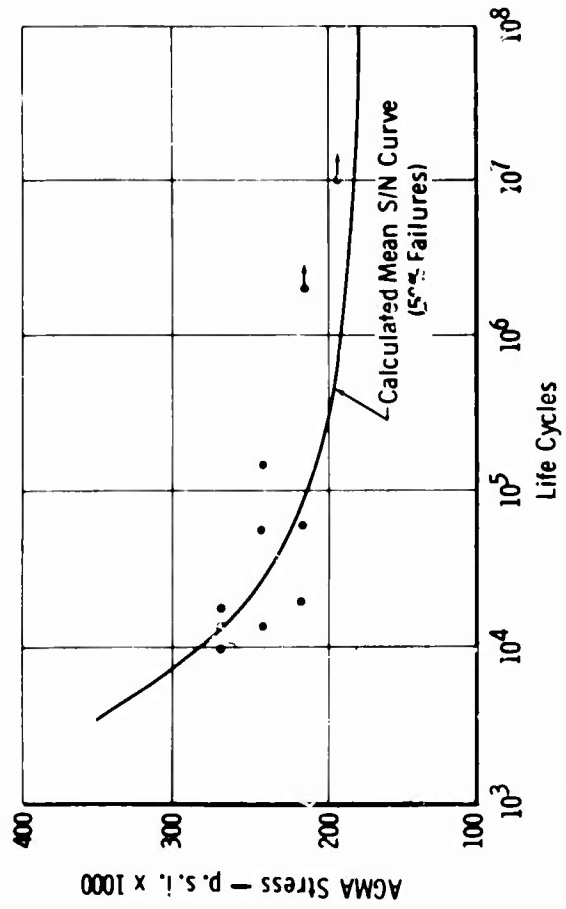


Figure 145. Fatigue Test Results—AGMA Stress Versus Life (EX-78782).
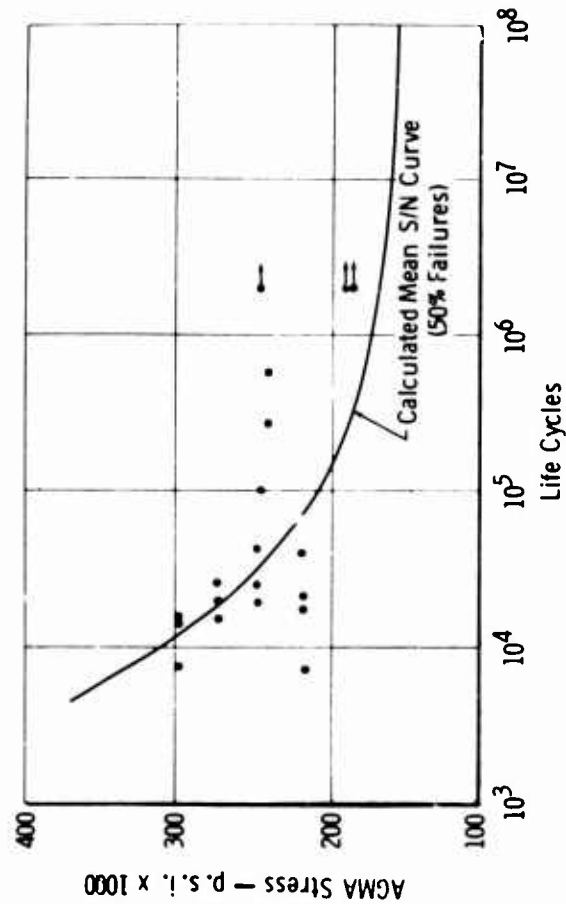


Figure 146. Fatigue Test Results—AGMA Stress Versus Life (EX-78783).

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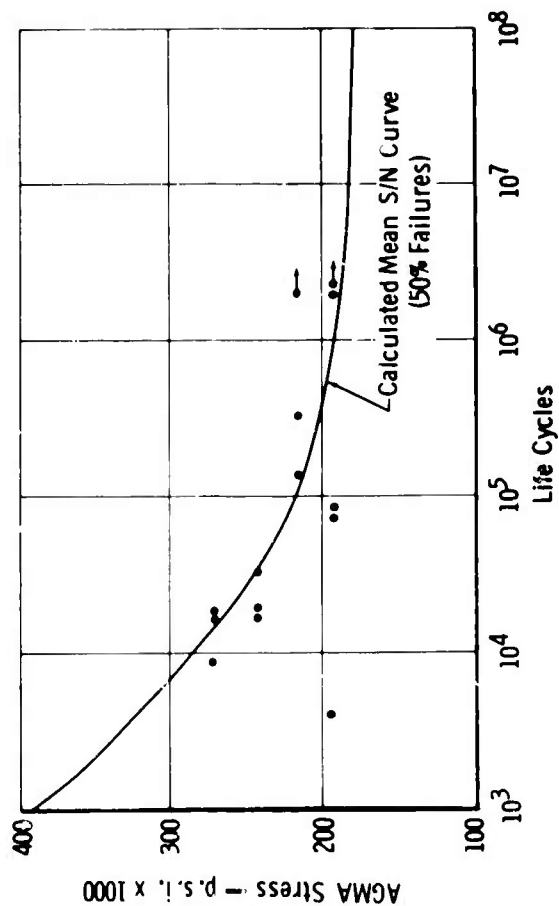
Figure 147. Fatigue Test Results—AGMA Stress Versus Life (EX-78784).

Figure 148. Fatigue Test Results—AGMA Stress Versus Life (EX-78785).
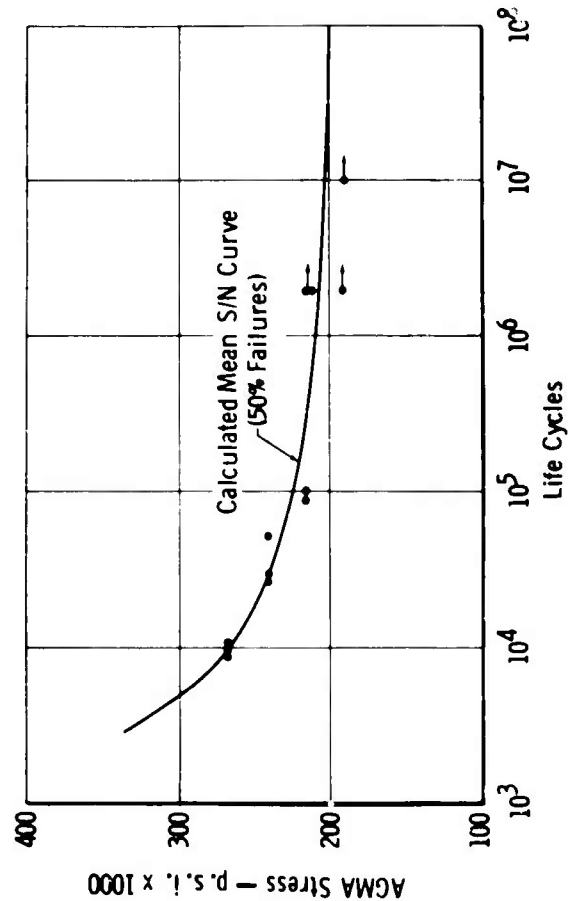
Figure 149. Fatigue Test Results—AGMA Stress Versus Life (EX-78786).
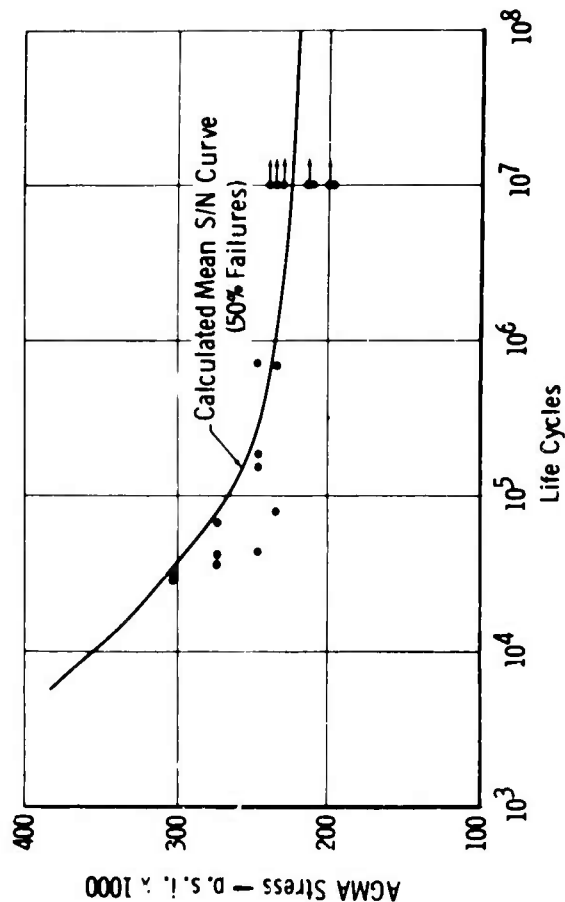
Figure 150. Fatigue Test Results—AGMA Stress Versus Life (EX-78787).

219
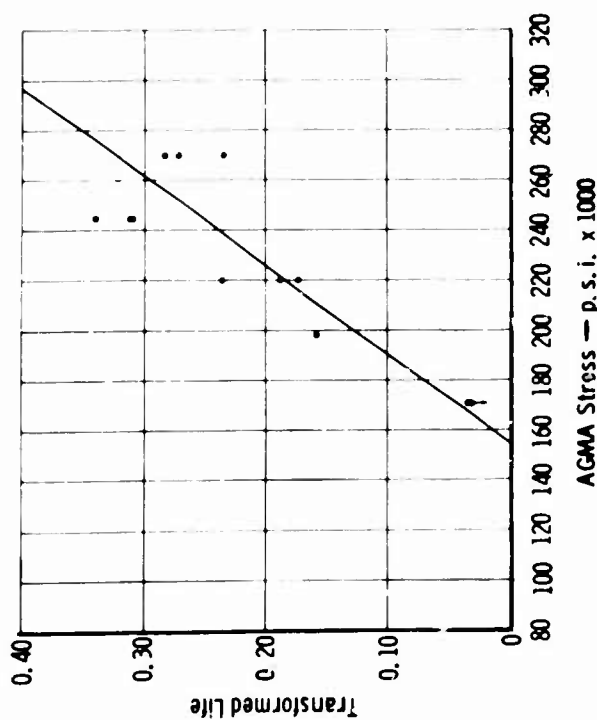
Figure 151. Fatigue Test Results—AGMA Stress Versus Transformed Life (EX-78772).



Figure 152. Fatigue Test Results—AGMA Stress Versus Transformed Life (EX-78773).
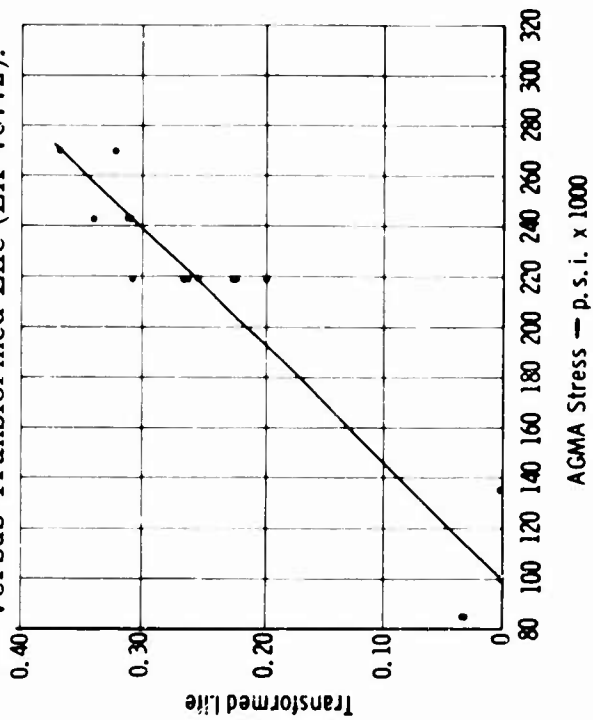


Figure 153. Fatigue Test Results—AGMA Stress Versus Transformed Life (EX-78774).
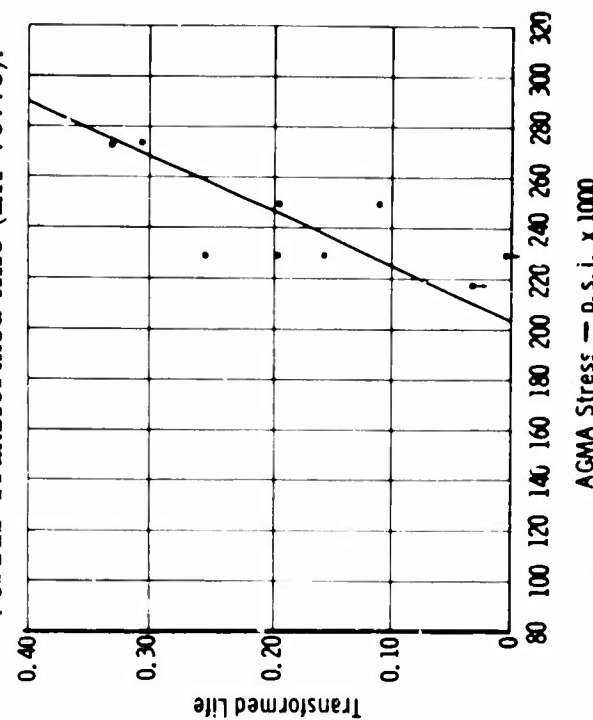


Figure 154. Fatigue Test Results—AGMA Stress Versus Transformed Life (EX-78775).

220
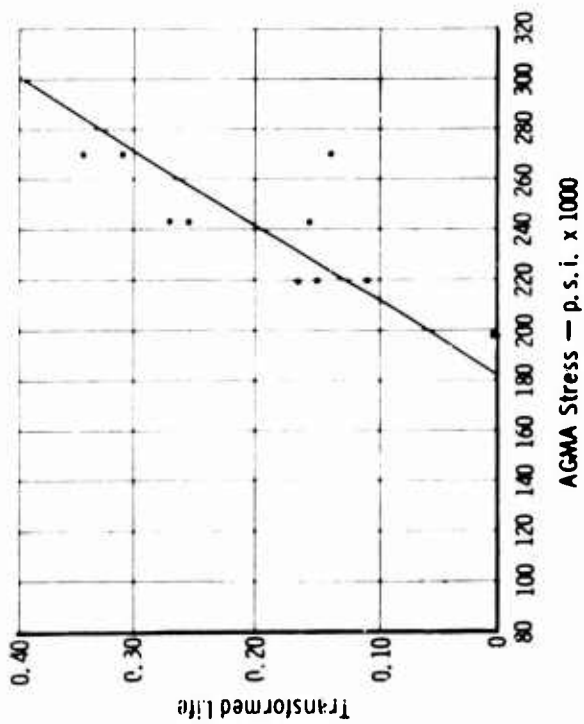
Figure 155. Fatigue Test Results–AGMA Stress Versus Transformed Life (EX-78776).



Figure 156. Fatigue Test Results–AGMA Stress Versus Transformed Life (EX-78777).
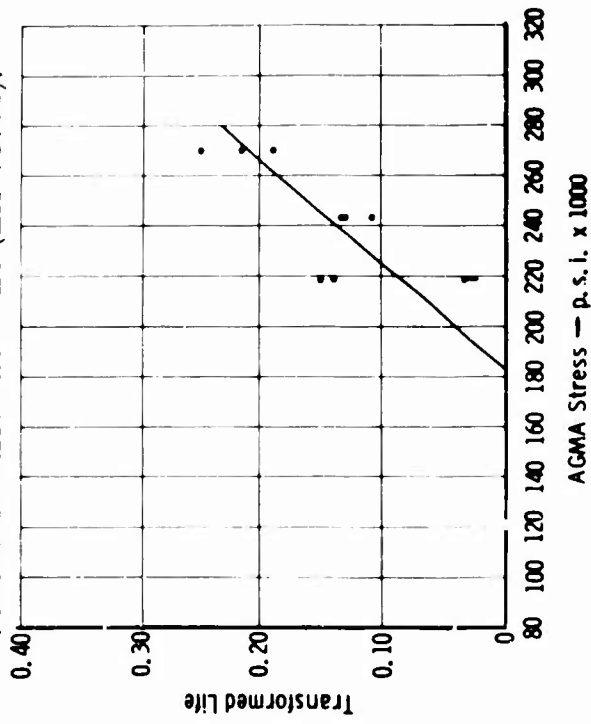


Figure 157. Fatigue Test Results–AGMA Stress Versus Transformed Life (EX-78778).
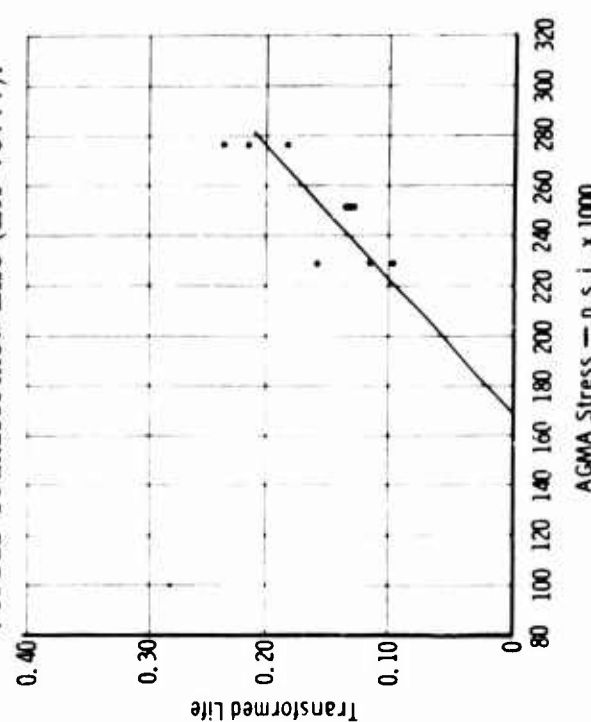


Figure 158. Fatigue Test Results–AGMA Stress Versus Transformed Life (EX-78779).

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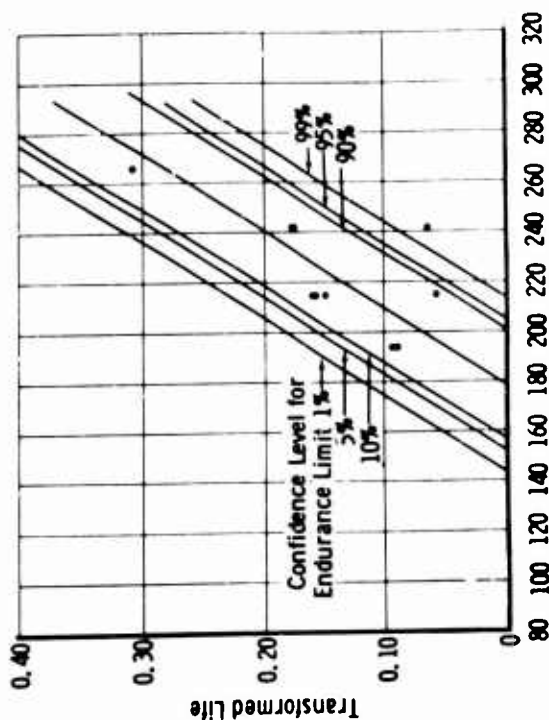
Figure 159. Fatigue Test Results–AGMA Stress Versus Transformed Life (EX-78780).

Figure 160. Fatigue Test Results–AGMA Stress Versus Transformed Life (EX-78781).
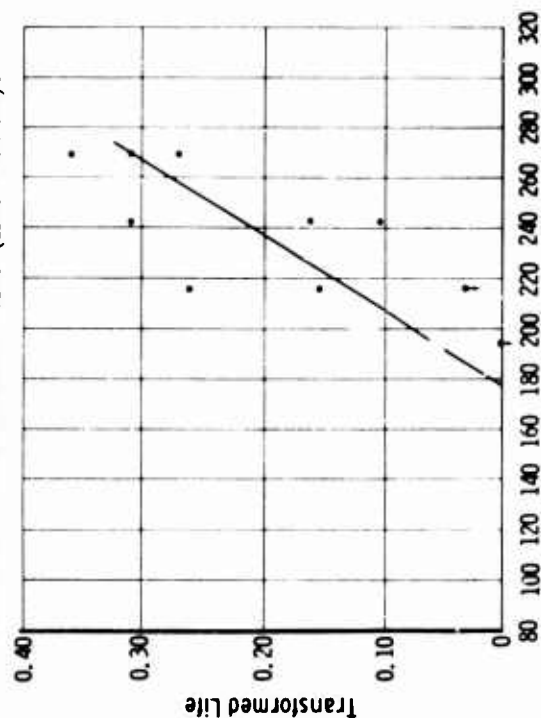
Figure 161. Fatigue Test Results–AGMA Stress Versus Transformed Life (EX-78782).
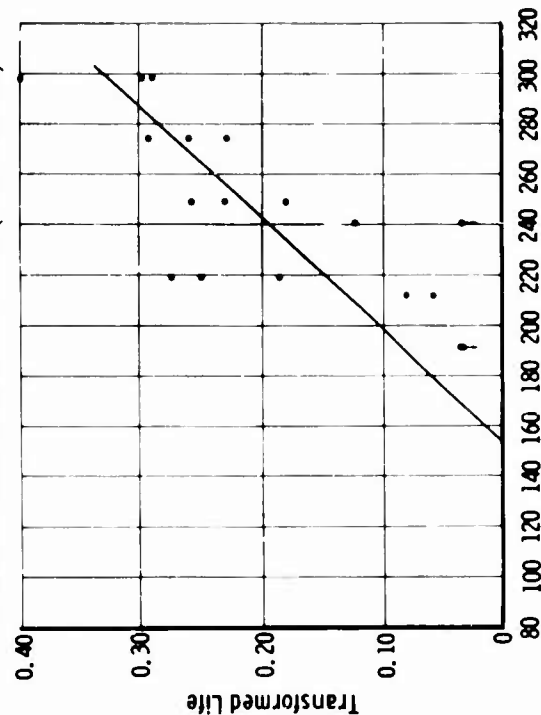
Figure 162. Fatigue Test Results–AGMA Stress Versus Transformed Life (EX-78783).

222
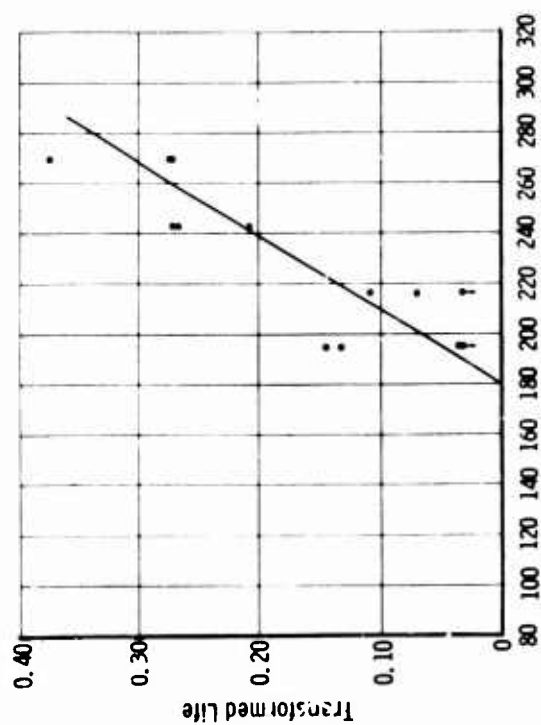
Figure 163. Fatigue Test Results—AGMA Stress
Versus Transformed Life (EX-78784).



Figure 164. Fatigue Test Results—AGMA Stress
Versus Transformed Life (EX-78785).



Figure 165. Fatigue Test Results—AGMA Stress
Versus Transformed Life (EX-78786).
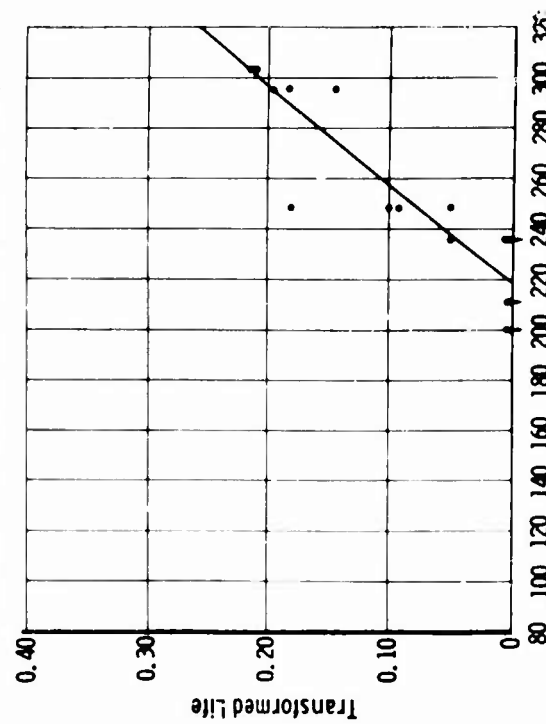


Figure 166. Fatigue Test Results—AGMA Stress
Versus Transformed Life (EX-78787).

223
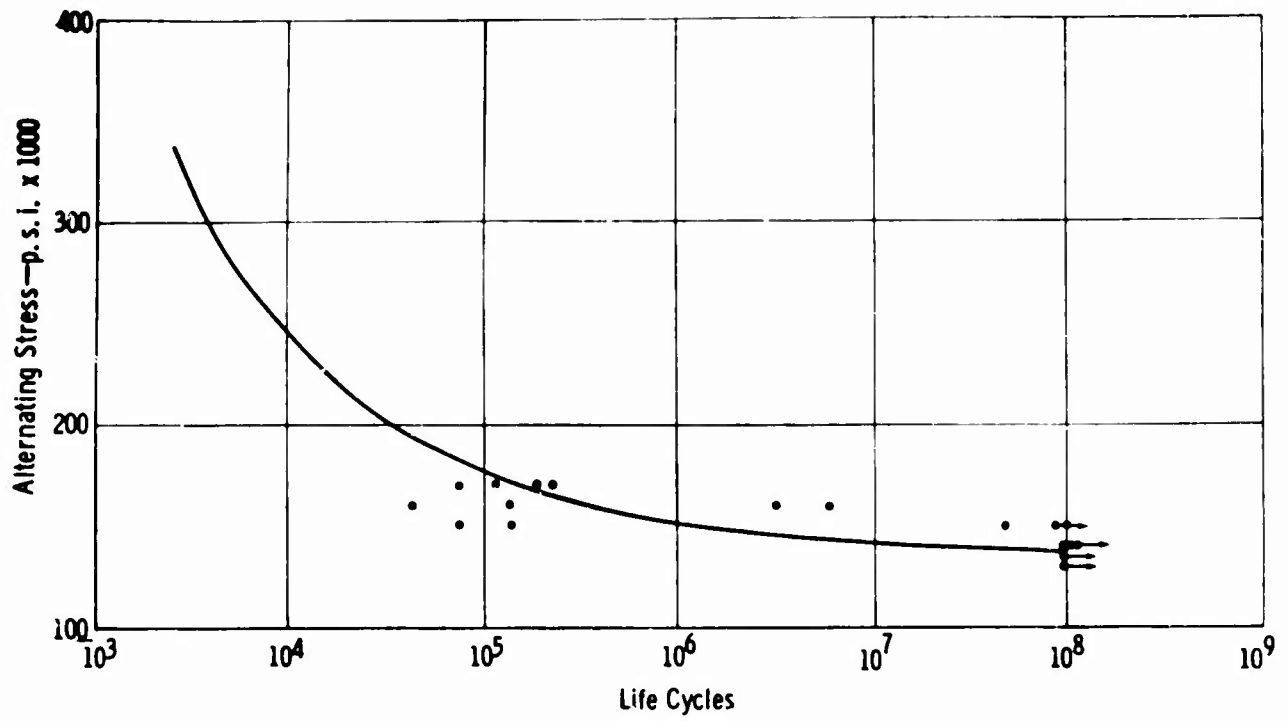
Figure 167.   Fatigue Test Gear Life Data (R. R. Moore).
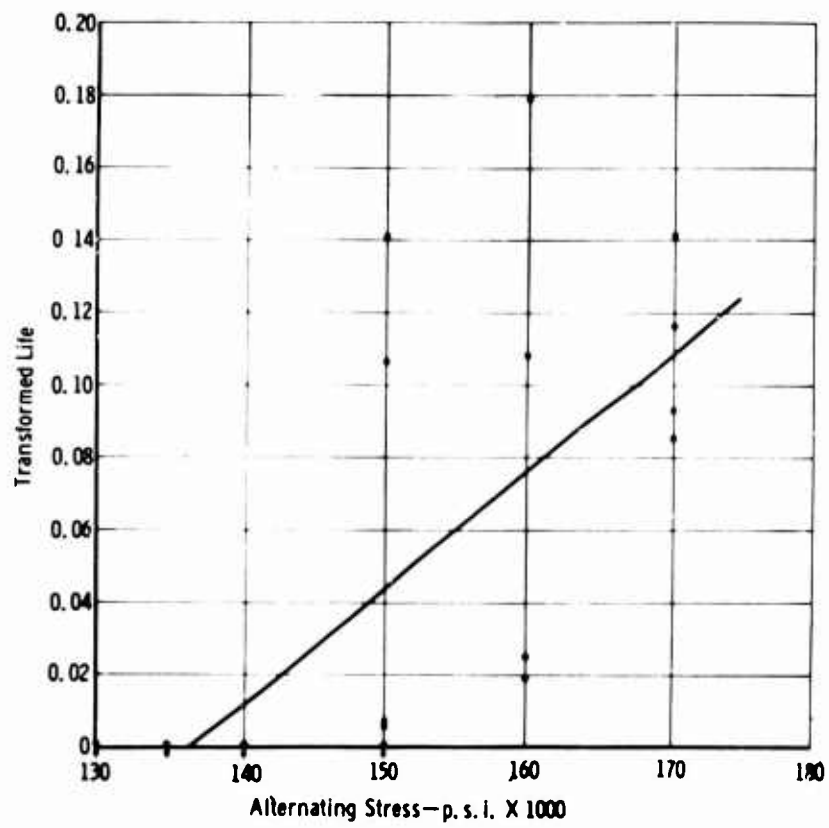


Figure 168.   Fatigue Test Gear Transformed Life Data (R. R. Moore).

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## DESCRIPTION OF COMPUTER PROGRAM

This appendix consists of a complete description of the computer program and includes the program equations, input data sheet, source program print-out, and a sample problem. The equations are given in both engineering and computer program terms.

### DESCRIPTION OF PROBLEM

Gear tooth bending strength is one of the major criteria in gear design. Gear tooth loading is cyclic in nature, therefore subjecting the material to fatigue. The critical section is close to the root diameter. Failure usually results in fracture of an entire tooth from the gear rim.

Calculation of gear tooth bending stress requires geometrically precise description of the root fillet contour and location of the critical section. The point of the involute tooth profile at which the transmitted load produces the maximum bending stress is also required. Knowledge of the mounting and operating conditions of the unit in which the gear is assembled is required to assess the increase in bending stress caused by misalignment, overloads, system dynamics, and centrifugal forces. Gear material ultimate strength and fatigue data must be known to convert the calculated stress to anticipated gear life.

The purpose of this program is to calculate gear tooth bending stress and gear life considering these factors.

### METHOD OF SOLUTION

The gear tooth geometry has been developed using basic formulas available in the literature. The hob dimensions have been used to generate in the program the trochoidal fillet contour resulting on a finished gear from some gear processing procedures. A true radius fillet is used when a shaped contour is specified in the program input. The program uses an iteration routine to inscribe a parabola (per Lewis construction) and to locate its tangent point with the root fillet contour. The Lewis dimensional parameters for the weakest section thus obtained are then calculated. These parameters are then used in the AGMA formula as given in AGMA 220.02 (Appendix VI herein) to calculate a bending stress. A hoop stress at the root diameter is also calculated. The AGMA temperature factor and factor of safety are applied to the bending and hoop stresses, which are then combined by use of a modified Goodman diagram. The modified Goodman diagram is based on an ultimate strength and S/N curve determined for the material used and the gear tooth designs tested; they may be easily changed within the program. A life is also determined from the modified Goodman diagram.

### COMPUTER TYPE AND PROGRAM LANGUAGE

The subject program is written in FORTRAN IV language for use on an IBM 7094 computer.

There must be four, five, or six cards per data set depending on data input for words 4 and 5 on Card 1. Data sets may be stacked. Computer running time will be approximately 0.1 minute per set of data.

## INPUT DATA

A sample input data form is shown in Figure 169. Each set of data requires four, five, or six cards. A description of the cards follows.

### Input Card 1

| Word | Column | Description |
|---|---|---|
| 1 | 1 - 5 | Number of teeth—Pinion. |
| 2 | 6 - 10 | Number of teeth—Gear. |
| 3 | 11 - 20 | Nonstandard center distance (blank if standard gear set). |
| 4 | 21 - 26 | This must be one of the following beginning in Column 21: SHAPED HOBBED |
| — | 27 - 29 | These spaces left blank. |
| 5 | 30 | This must be one of the following in Column 30: 0—if pinion is hobbed 1—if gear is hobbed 2—if both pinion and gear are hobbed Blank--if "SHAPED" is in Column 21 through 26 |
| 6 | 31 - 40 | Horsepower. |
| 7 | 41 - 50 | r.p.m.—Pinion. |
| 8 | 51 - 55 | Density—pounds/cubic inch. |
| 9 | 56 - 60 | Temperature factor. |
| 10 | 61 - 65 | Safety factor. |
| 11 | 66 - 70 | Load distribution factor. |

### Input Card 2

| Word | Column | Description |
|---|---|---|
| 1 | 1 - 10 | Pressure angle at the standard pitch diameter—degrees. |
| 2 | 11 - 20 | Diametral pitch at the standard pitch diameter. |
| 3 | 21 - 25 | Backlash—minimum. |
| 4 | 26 - 30 | Backlash—maximum. |
| 5 | 31 - 40 | Arc or chordal tooth thickness at the standard pitch diameter—minimum (pinion). |
| 6 | 41 - 50 | Arc or chordal tooth thickness at the standard pitch diameter—maximum (pinion). |
| 7 | 51 - 60 | Arc or chordal tooth thickness at the standard pitch diameter—minimum (gear). |
| 8 | 61 - 70 | Arc or chordal tooth thickness at the standard pitch diameter—maximum (gear). |
| — | 71 | This space is left blank. |
| 9 | 72 | This must be one of the following in Column 72: 0—if Columns 31 through 70 are arc tooth thickness 1—if Columns 31 through 70 are chordal tooth thickness |

### Input Card 3

| Word | Column | Description |
|---|---|---|
| 1 | 1 - 10 | Outside diameter—minimum (pinion). |
| 2 | 11 - 20 | Outside diameter—maximum (pinion). |
| 3 | 21 - 30 | Outside diameter—minimum (gear). |
| 4 | 31 - 40 | Outside diameter—maximum (gear). |

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Figure 169. Sample Input Data Form.

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| Word | Column | Description |
|------|--------|-------------|
| 5 | 41 - 50 | **Face width**—minimum (pinion). |
| 6 | 51 - 60 | **Face width**—minimum (gear). |
| 7 | 61 - 65 | **Maximum** tip break (pinion). |
| 8 | 66 - 70 | **Maximum** tip break (gear). |

## Input Card 4

| | | |
|------|--------|-------------|
| 1 | 1 - 10 | **Root diameter**—minimum (pinion). |
| 2 | 11 - 20 | **Root diameter**—maximum (pinion). |
| 3 | 21 - 30 | **Root diameter**—minimum (gear). |
| 4 | 31 - 40 | **Root diameter**—maximum (gear). |
| 5 | 41 - 45 | Fillet radius—minimum (pinion) (blank if pinion is hobbed). |
| 6 | 46 - 50 | Fillet radius—minimum (gear) (blank if gear is hobbed). |
| 7 | 51 - 55 | Maximum undercut (pinion) (blank if pinion is hobbed). |
| 8 | 56 - 60 | Maximum undercut (gear) (blank if gear is hobbed). |
| 9 | 61 - 65 | **Overload factor.** |
| 10 | 66 - 70 | **Dynamic factor.** |

## Input Card 5

This card is needed only when words 4 and 5 of Card 1 are given as "HOBBED" and "0" or "2," respectively. This card is for PINION only. See Figure 170.

H LEAD - HOB LEAD
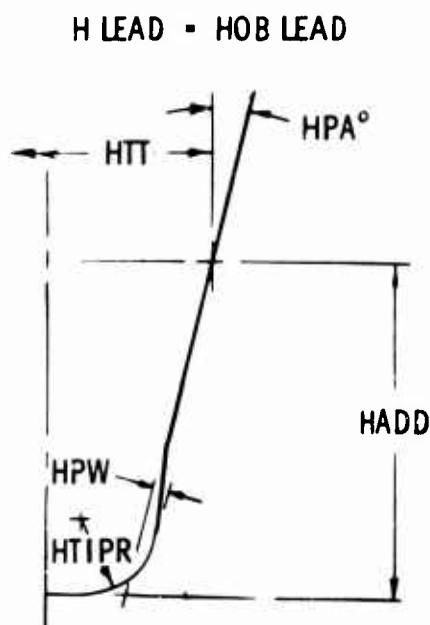


Figure 170. Standard or Protuberance Hob Form for Input.

| Word | Column | Description |
|------|--------|-------------|
| 1 | 1 - 10 | Hob tooth thickness. |
| 2 | 11 - 20 | Hob addendum. |
| 3 | 21 - 30 | Hob lead. |
| 4 | 31 - 40 | Hob pressure angle—degrees. |
| 5 | 41 - 50 | Hob tip radius—inches. |
| 6 | 51 - 6C | HPW. (See Figure 170.) |

### Input Card 6

This card is needed only when words 4 and 5 of Card 1 are given as "HOBBED" and "1" or "2," respectively. This card is for GEAR only and is the same format as input Card 5.

### PROGRAM EQUATIONS

Computer program input symbols in both engineering (AGMA) and program terms are listed as follows.

| AGMA | Program | Definition |
|------|---------|------------|
| NP | ANP | Number of teeth—pinion. |
| NG | ANG | Number of teeth—gear. |
| $B_{MI}$ | BMIN | Backlash—minimum. |
| $B_{MA}$ | BMAX | Backlash—maximum. |
| — | BRKP | Maximum tip break—pinion. |
| -- | BRKG | Maximum tip break—gear. |
| C | CSTDIN | Standard center distance. |
| $C_X$ | CNSTD | Nonstandard center distance. |
| — | CODE | See input fillout. |
| — | CUTTER | See input fillout. |
| $D_{OMA}$ | DOGMA | Outside diameter—maximum (gear). |
| $D_{OMI}$ | DOGMI | Outside diameter—minimum (gear). |
| $d_{OMA}$ | DOPMA | Outside diameter—maximum (pinion). |
| $d_{OMI}$ | DOPMI | Outside diameter—minimum (pinion). |
| $DR_{MA}$ | DRGMA | Root diameter—maximum (gear). |
| $DR_{MI}$ | DRGMI | Root diameter—minimum (gear). |
| $dR_{MA}$ | DRPMA | Root diameter—maximum (pinion). |
| $dR_{MI}$ | DRPMI | Root diameter—minimum (pinion). |
| $FG_{MI}$ | FMING | Face width—minimum (gear). |
| $Fp_{MI}$ | FMINP | Face width—minimum (pinion). |
| HP | HORSES | Horsepower. |
| — | L | See input fillout. |
| $K_m$ | KM | Load distribution factor. |
| $K_O$ | KO | Overload factor. |
| $K_R$ | KR | Safety factor. |
| $K_T$ | KT | Temperature factor. |
| $K_V$ | KV | Dynamic factor. |
| $n_P$ | RPMP | r.p.m.—pinion. |
| | | Arc or chordal tooth thickness |
| $t_{GMA}$ or $tc_{GMA}$ | TGMAS | Maximum—gear. |
| $t_{GMI}$ or $tc_{GMI}$ | TGMIS | Minimum—gear. |

| AGMA | Program | Definition |
|------|---------|------------|
| $t_{pMA}$ or $tcp_{MA}$ | TPMAS | Maximum—pinion. |
| $t_{pMI}$ or $tcp_{MI}$ | TPMIS | Minimum—pinion. |
| $r_{fGMI}$ | RFMIG | True root fillet radius—gear. |
| $r_{fpMI}$ | RFMIP | True root fillet radius—pinion. |
| — | UCG | Maximum undercut—gear. |
| — | UCP | Maximum undercut—pinion. |
| $a_C$ | HADD | Hob addendum. |
| $L_C$ | HLEAD | Hob lead. |
| $\phi_C$ | HPA | Hob pressure angle. |
| — | HPW | Hob protuberance. |
| $r_T$ | HTIPR | Hob tip radius. |
| $t_c$ | HTT | Hob tooth thickness. |

The computer program equations in both engineering (AGMA) and program terms follow. The basic geometric equations for gear teeth can be obtained or developed from textbooks.

| AGMA | Program |
|------|---------|
| $Pd_x = \dfrac{Np + NG}{2 \times C_x}$ | $PDX = \dfrac{ANP + ANG}{2 \times CNSTD}$ |
| $mg = \dfrac{NG}{Np}$ | $AMG = \dfrac{ANG}{ANP}$ |
| $Rmg = \dfrac{Np}{NG}$ | $RMG = \dfrac{ANP}{ANG}$ |
| $dp = \dfrac{Np}{Pnd}$ | $DP = \dfrac{ANP}{PND}$ |
| $db = dp \times \cos \phi_\eta$ | $DBP = DP \times FNCO$ |
| $d_x = \dfrac{Np}{Pd_x}$ | $DXP = \dfrac{ANP}{PDX}$ |
| $D_G = \dfrac{NG}{Pnd}$ | $DG = \dfrac{ANG}{PND}$ |
| $Db = D_G \times \cos \phi_\eta$ | $DBG = DG \times FNCO$ |
| $D_x = \dfrac{NG}{Pd_x}$ | $DXG = \dfrac{ANG}{PDX}$ |
| $d_{ODB} = d_{OMI} - 2 \times BRKp$ | $DODBP = DOPMI - 2 \times BRKP$ |
| $D_{ODB} = D_{OMI} - 2 \times BRKG$ | $DODBG = DOGMI - 2 \times BRKG$ |
| $\epsilon_{ECP} = \left[\left(\dfrac{d_{ODB}}{db}\right)^2 - 1\right]^{1/2}$ | $EECP = \left[\left(\dfrac{DODBP}{DBP}\right)^2 - 1\right]^{1/2}$ |

| AGMA | Program |
|---|---|

$$\epsilon_{BCG} = \left[\left(\frac{DODB}{Db}\right)^2 - 1\right]^{1/2} \qquad EBCG = \left[\left(\frac{DODBG}{DBG}\right)^2 - 1\right]^{1/2}$$

$$\epsilon_{BCP} = (TAN\,\phi_x(mg+1)) - (\epsilon_{BCG} \times mg) \qquad EBCP = (FXTA(AMG+1)) - (EBCG \times AMG)$$

$$\epsilon_{ECG} = (TAN\,\phi_x(Rmg+1)) - (\epsilon_{ECP} \times Rmg) \qquad EECG = (FXTA(RMG+1)) - (EECP \times RMG)$$

$$\epsilon_{BSTCP} = \epsilon_{ECP} - \frac{2\pi}{NP} \qquad EBSP = EECP - \frac{2 \times PI}{ANP}$$

$$\epsilon_{ESTCP} = \epsilon_{BCP} + \frac{2\pi}{NP} \qquad EESP = EBCP + \frac{2 \times PI}{ANP}$$

$$\epsilon_{BSTCG} = \epsilon_{ECG} + \frac{2\pi}{NG} \qquad EBSG + EECG + \frac{2 \times PI}{ANG}$$

$$\epsilon_{ESTCG} = \epsilon_{BCG} - \frac{2\pi}{NG} \qquad EESG = EBCG - \frac{2 \times PI}{ANG}$$

$$\epsilon_{dcMA} = \left[\left(\frac{dOMA}{d_b}\right)^2 - 1\right]^{1/2} \qquad E_{OPMA} = \left[\left(\frac{DOPMA}{DBP}\right)^2 - 1\right]^{1/2}$$

$$\epsilon_{DOMA} = \left[\left(\frac{DOMA}{D_b}\right)^2 - 1\right]^{1/2} \qquad E_{OGMA} = \left[\left(\frac{DOGMA}{DBG}\right)^2 - 1\right]^{1/2}$$

$$d_{BC} = \left[\epsilon_{BCP}^2 + 1\right]^{1/2} db \qquad DBCP = \left[EBCP^2 + 1\right]^{1/2} \times DBP$$

$$d_{BSTC} = \left[\epsilon_{BSTCP}^2 + 1\right]^{1/2} db \qquad DBSP = \left[EBSP^2 + 1\right]^{1/2} \times DBP$$

$$d_{ESTC} = \left[\epsilon_{ESTCP}^2 + 1\right]^{1/2} db \qquad DESP = \left[EESP^2 + 1\right]^{1/2} \times DBP$$

$$d_{EC} = \left[\epsilon_{ECP}^2 + 1\right]^{1/2} db \qquad DECP = \left[EECP^2 + 1\right]^{1/2} \times DBP$$

$$D_{BC} = \left[\epsilon_{BCG}^2 + 1\right]^{1/2} Db \qquad DBCG = \left[EBCG^2 + 1\right]^{1/2} \times DBG$$

$$D_{BSTC} = \left[\epsilon_{BSTCG}^2 + 1\right]^{1/2} Db \qquad DBSG = \left[EBSG^2 + 1\right]^{1/2} \times DBG$$

$$D_{ESTC} = \left[\epsilon_{ESTCG}^2 + 1\right]^{1/2} Db \qquad DESG = \left[EESG^2 + 1\right]^{1/2} \times DBG$$

$$D_{EC} = \left[\epsilon_{ECG}^2 + 1\right]^{1/2} Db \qquad DECG = \left[EECG^2 + 1\right]^{1/2} \times DBG$$

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|  |  |
|---|---|
| <div align="center">**AGMA**</div> | <div align="center">**Program**</div> |

$$m_{N_{max}} = NG \, (\epsilon_{OG} - TAN\phi_x) + \Big/ \atop (Np \, (\epsilon_{OP} - TAN\phi_x)) \Big/ 2\pi$$

$$AMP_{MA} = ANG \, (EOGMA - FXTA) + \atop ANP \, (EOPMA - FXTA) \Big/ 2\pi$$

$$m_{N_{min}} = NG \, (\epsilon_{BCG} - TAN\phi_x) + \Big/ \atop (NP \, (\epsilon_{ECP} - TAN\phi_x)) \Big/ 2\pi$$

$$AMP_{MI} = ANG \, (EBCG - FXTA) + \atop ANP \, (EECP - FXTA) \Big/ 2\pi$$

See Figure 171.

$$SIN \, (AN) = \frac{0.5 \, t_c}{0.5 \, D}$$

$$AN = \frac{0.5 \times TPMIS}{0.5 \times DP}$$

$$\widehat{AN} = ARC \, TAN \left( \frac{AN}{\sqrt{1 - AN^2}} \right)$$

$$AN = ATAN \left( \frac{AN}{\sqrt{1 - AN^2}} \right)$$

$$t = \widehat{AN} \times D$$

$$TPMIS = AN \times DP$$

$$t_x = D_x \left[ \left( \left( \frac{t}{D} \right) + INV\phi \right) - INV\phi_x \right]$$

$$TPMIN = D \times P \left[ \left( \left( \frac{TPMIS}{DP} \right) + ZF \right) - ZFX \right]$$
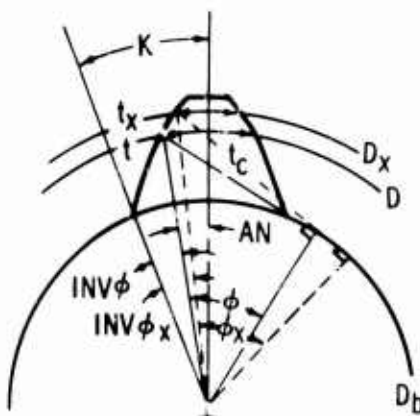


Figure 171. Arc and Chordal Tooth Thickness.

|                | AGMA                                                    |                | Program                                             |
|----------------|---------------------------------------------------------|----------------|-----------------------------------------------------|

$$\cos \phi_x = \frac{Db}{D_x}$$

$$F = \frac{DB}{DIA(I)}$$

$$\hat{\phi}_x = \text{ARC TAN} \left( \frac{\sqrt{1 - \phi_x^2}}{\phi_x} \right)$$

$$FRA(I) = \text{ATAN} \left( \frac{\sqrt{1 - F^2}}{F} \right)$$

$$\text{INV} \, \phi_x = \text{TAN} \, (\hat{\phi}_x) - \hat{\phi}_x$$

$$ZF(I) = FTA(I) - FRA(I)$$

$$K = \frac{t}{D_x} + \text{INV} \, \phi_x$$

$$PK = \frac{TPMIN}{DXP} + ZFX$$

$$F = \text{TAN} \, (\phi) - K$$

$$F(I) = FPTA(I) - PK$$

$$D_V = \frac{Db}{\cos (F)}$$

$$DVP(I) = \frac{DBP}{\cos(F(I))}$$

Basic Hob Data (See Figure 172.)

<div align="center">Program</div>

$$TSA = \frac{\pi}{N}$$

$$DHPA = N \times \frac{HLEAD}{\pi}$$

$$HADDN = 0.5 \, (DHPA - D_R)$$

$$HPAR = 0.017453293 \times HPA$$

$$HTTN = HTT + 2 \, (HADDN - HADD) \, \text{TAN} \, (HPAR)$$

$$HTTR = 0.5 \times HTT - HADD \times \text{TAN} \, (HPAR)$$

$$HA = HTTR - \frac{HTIPR - HPW}{\cos (HPAR)}$$

$$HRCTRX = HA + HTIPR \times \text{TAN} \, (HPAR)$$

$$RHPA = 0.5 \, DHPA$$

$$HRCTRP = HADDN - HTIPR$$

$$\widehat{HPCA} = \text{ARC TAN} \left( \frac{HRCTRX}{HRCTRP} \right)$$

$$HYP = \sqrt{HRCTRX^2 + HRCTRP^2}$$

Wrap pitch line of hob around gear pitch circle by equal increments and calculate path of hob tip radius center. See Figure 173.
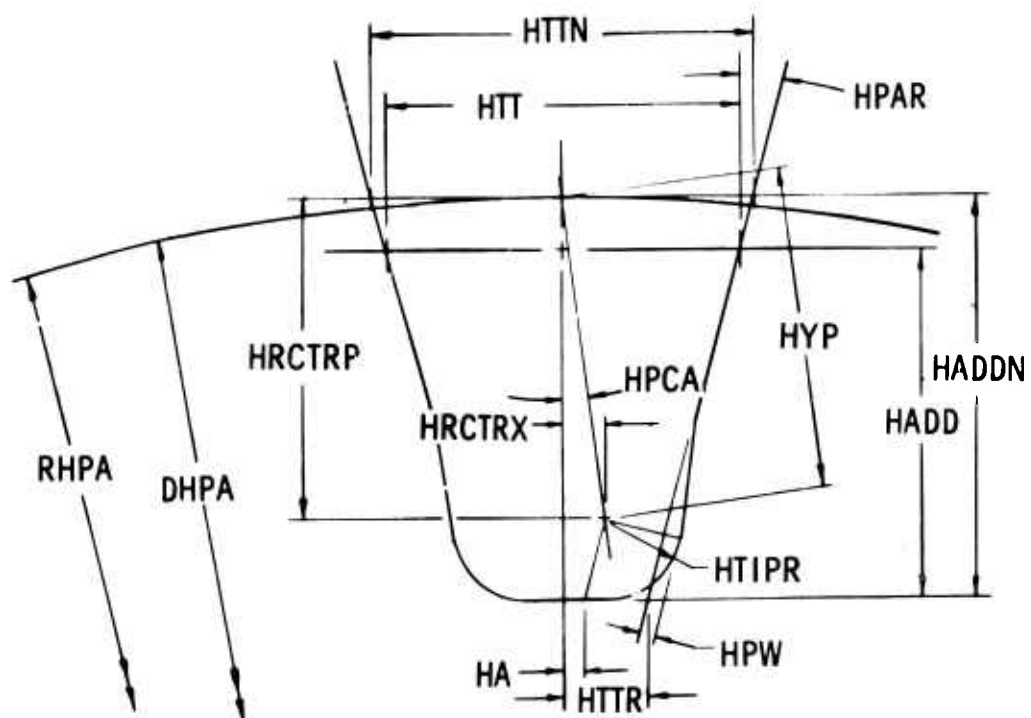
Figure 172. Standard or Protuberance Hob Form for Calculation.
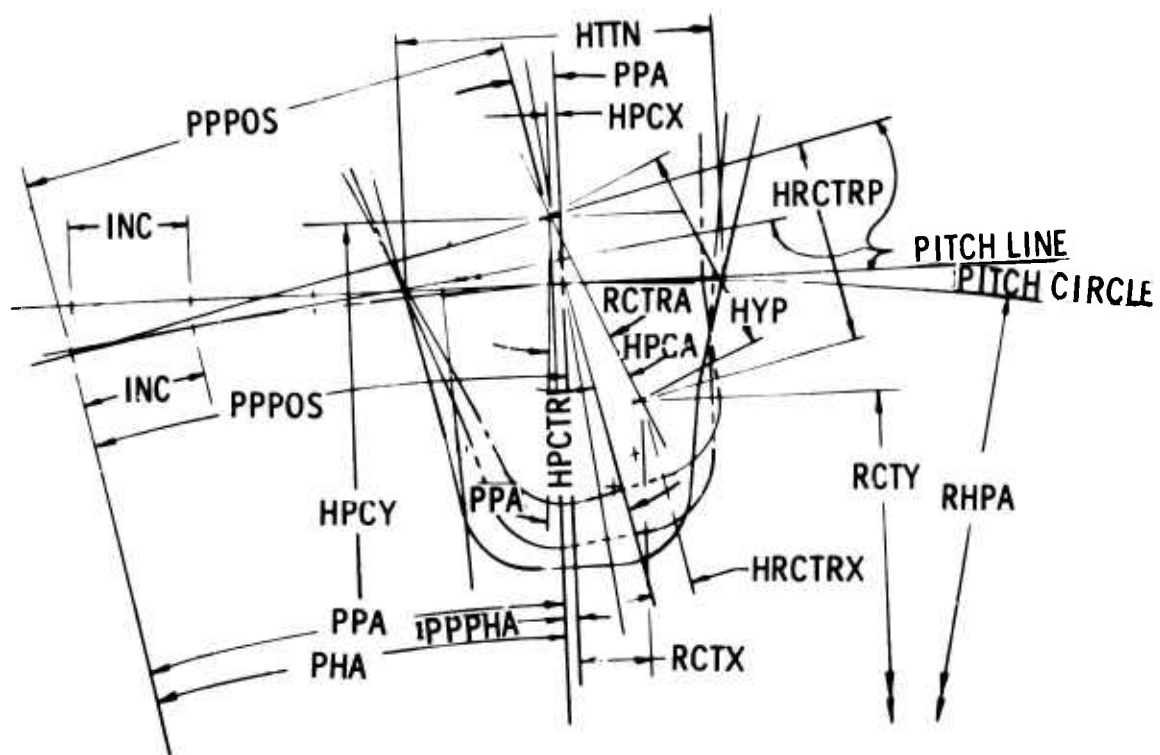


Figure 173. Tooth Generation by Hob.

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## Program

INC = 0,1 × HTTN          (increment of change)

PPPOS = 0                 (pitch point position—first time through increase PPPOS by increments each time)

$$PPA = \frac{PPPOS}{RHPA}$$

$$HPCTR = \sqrt{PPPOS^2 + RHPA^2}$$

$$\overset{\frown}{PHA} = ARC\ TAN\left(\frac{PPPOS}{RHPA}\right)$$

PPPHA = PPA - PHA

HPCX = HPCTR × SIN (PPPHA)

HPCY = HPCTR × COS (PPPHA)

RCTRA = HPCA + PPA

RCTX = HYP × SIN (RCTRA) - HPCX

RCTY = HPCY - HYP × COS (RCTRA)

Calculate points where hob tip radius is making final cut in fillet of gear. See Figure 174.
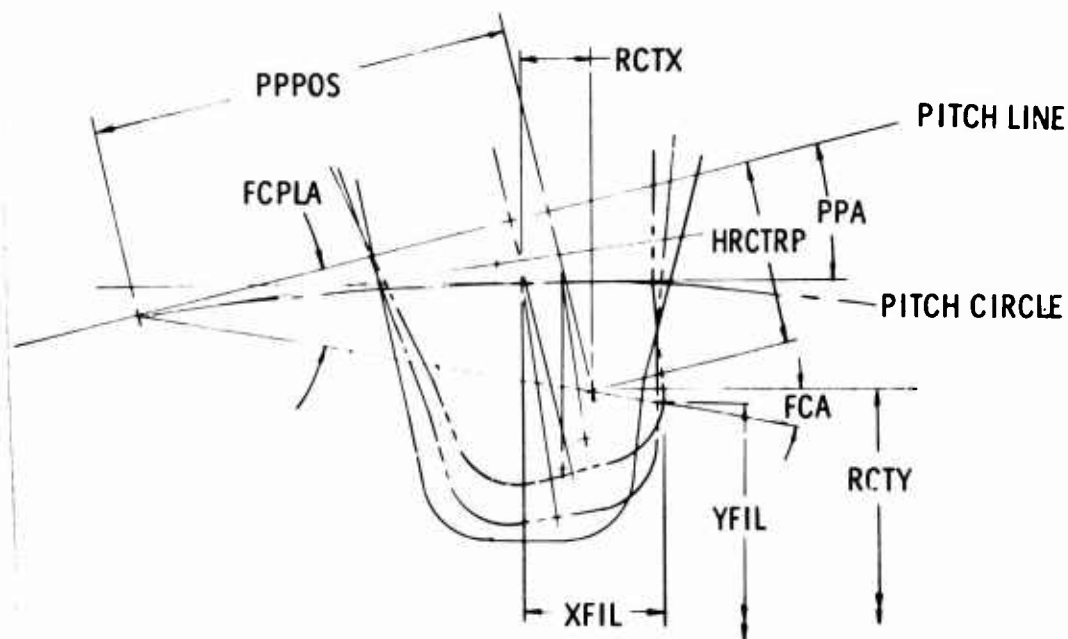


Figure 174.   Fillet Generation by Hob.

$$\widehat{\text{FCPLA}} = \text{ARC TAN} \left(\frac{\text{HRCTRP}}{\text{PPPOS}}\right)$$

$$\text{FCA} = \text{FCPLA} - \text{PPA}$$

$$\text{XFIL} = \text{RCTX} + \text{HTIPR} \times \text{COS (FCA)}$$

$$\text{YFIL} = \text{RCTY} - \text{HTIPR} \times \text{SIN (FCA)}$$

Convert location of fillet points from center of tooth space to center of gear tooth. See Figure 175.

$$\widehat{\text{FSA}} = \text{ARC TAN} \left(\frac{\text{XFIL}}{\text{YFIL}}\right)$$

$$\text{FTA} = \text{TSA} - \widehat{\text{FSA}}$$

$$\text{RFIL} = \sqrt{\text{XFIL}^2 + \text{YFIL}^2}$$

$$\text{XTFIL} = \text{RFIL} \times \text{SIN (FTA)}$$

$$\text{YTFIL} = \text{RFIL} \times \text{COS (FTA)}$$

Find parabola for evaluating bending stress. See Figure 176.
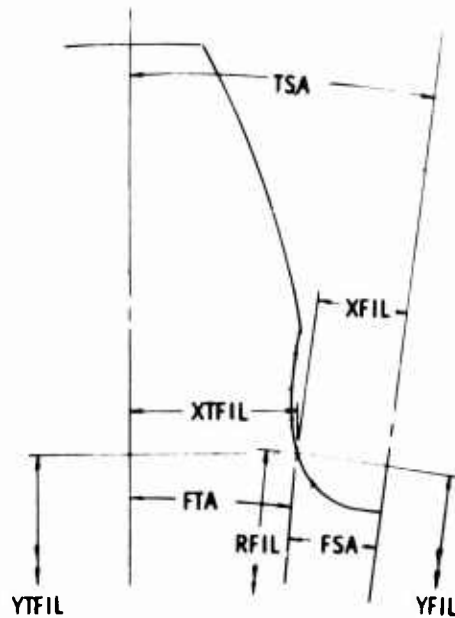


Figure 175. Generated Tooth Fillet.
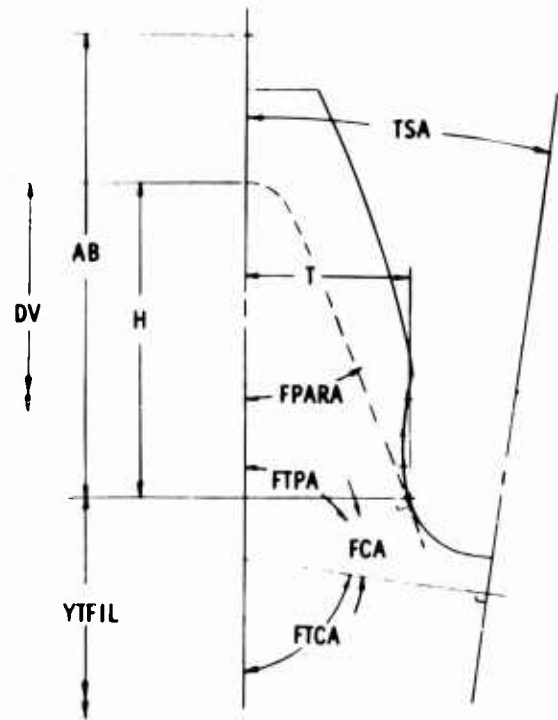
Figure 176. Trochoidal Fillet
Inscribed Lewis Parabola.

FTCA $= \frac{\pi}{2} -$ TSA

FTPA $= \pi - $ (FTCA + FCA)

FPARA $= \frac{\pi}{2} -$ FTPA

AB $=$ T $\times$ TAN (FPARA)

H $=$ 0.5 DV - YTFIL

Reiterate for new T, H, and YTFIL values until AB = 2H is satisfied.

Find the radius of curvature of generated fillet tangent to parabola. See Figure 177.

SIDEA $=$ YFIL - (RHPA - HADDN)

HYPA $= \dfrac{\text{SIDEA}}{\text{COS (FCA)}}$

ANGLEA $= 0.5 \left( \left( \frac{\pi}{2} \right) + \text{FCA} \right)$

FILR $=$ HYPA $\times$ TAN (ANGLEA)
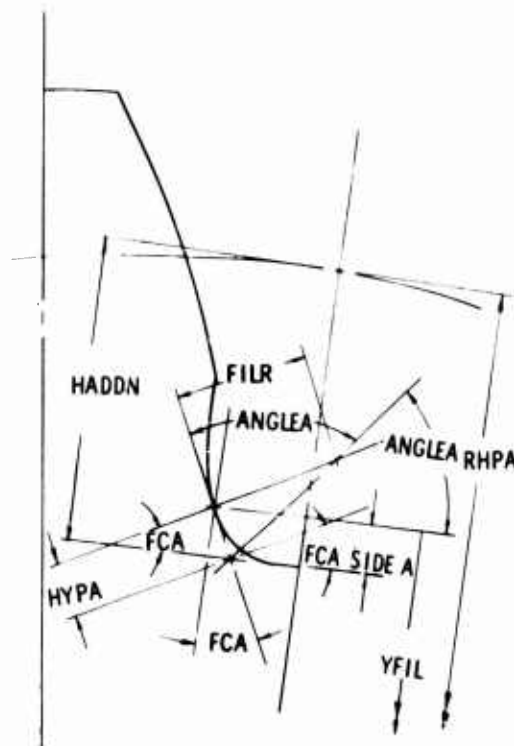


Figure 177. Radius of Curvature at Weakest Section.

Find X value from parabola and diameter of the weakest section of tooth. See Figure 178.

$$\text{ANGLED} = \text{ARC TAN} \left(\frac{T}{H}\right)$$

$$\text{ADJ} = \frac{T}{\text{SIN (ANGLED)}}$$

$$\text{XDIM} = \frac{\text{ADJ}}{\text{COS (ANGLED)} - H}$$

$$\text{DW} = 2\sqrt{T^2 + \text{YTFIL}^2}$$

Find coordinates to center of true fillet radius. See Figures 179 and 180

$$H = \frac{DR}{2} + RF$$

When $\frac{DB}{2} \leq H$, then (Figure 179):

$$\text{CPR} = \frac{0.5\,DB}{H}$$
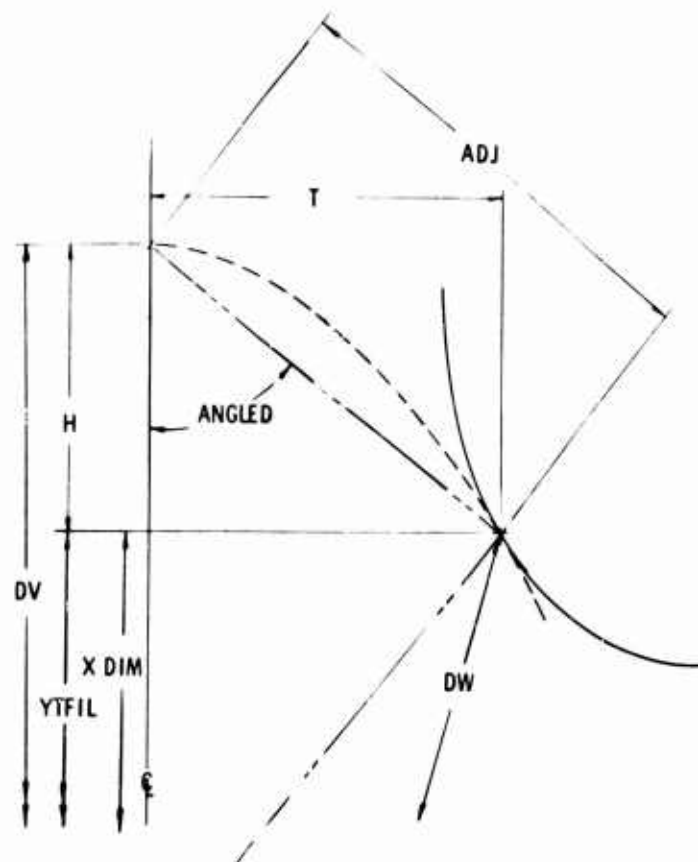


Figure 178. Diameter of Weakest Section and Lewis $Y$ Value.

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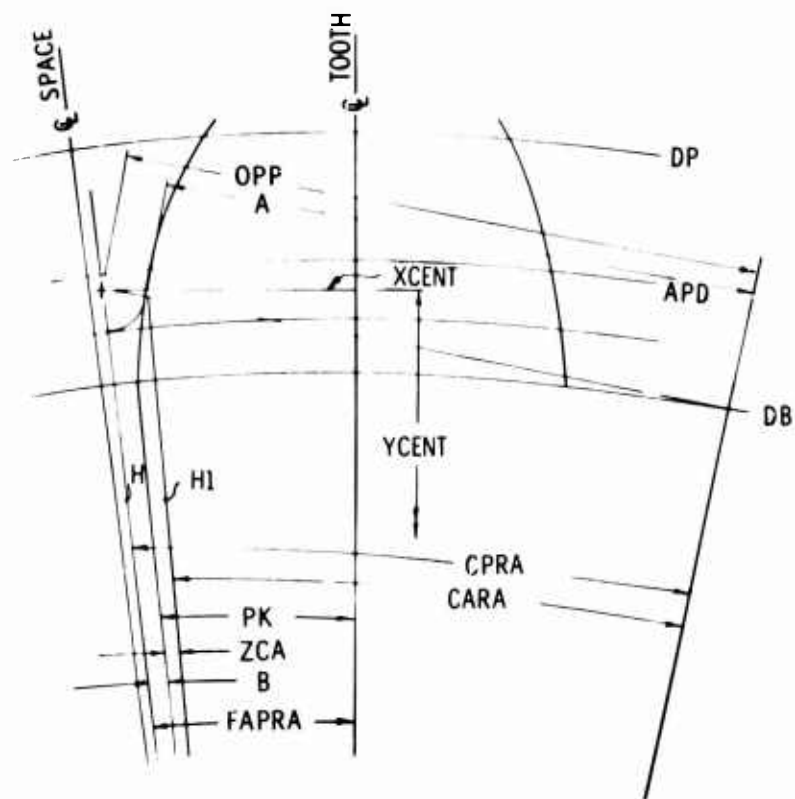
Figure 179. Coordinates at Center of True Fillet Radius—Base Circle Below Root Diameter.
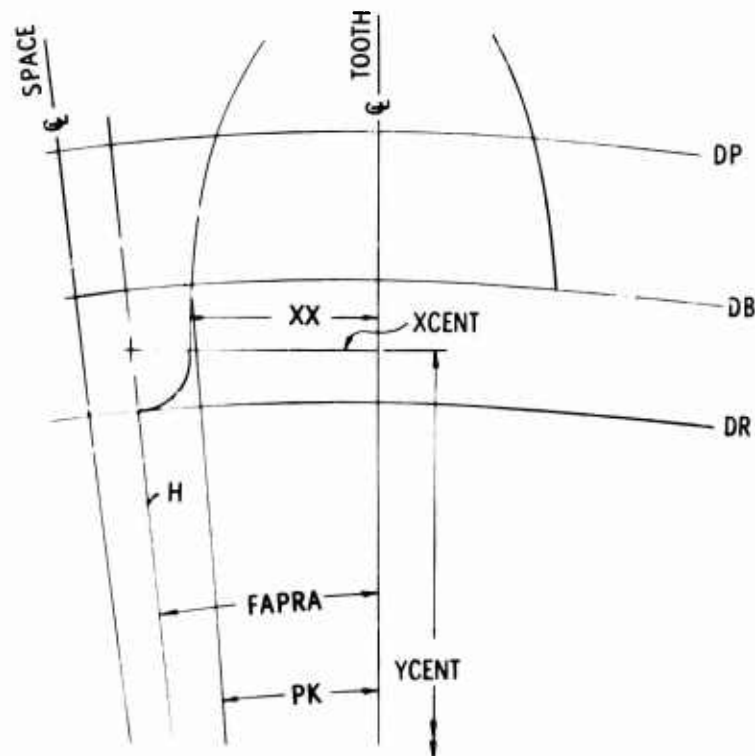


Figure 180. Coordinates at Center of True Fillet Radius—Base Circle Above Root Diameter.

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$$\text{CPRA} = \text{ARC TAN}\left(\frac{\sqrt{1 - CPR^2}}{CPR}\right)$$

$$\text{OPP} = \sqrt{H^2 - (0.5\,DB)^2}$$

$$A = OPP - RF$$

$$H1 = \sqrt{A^2 + (0.5\,DB)^2}$$

$$CA = \frac{0.5\,DB}{H1}$$

$$\text{CARA} = \text{ARC TAN}\left(\frac{\sqrt{1 - CA^2}}{CA}\right)$$

$$ZCA = \text{TAN (CARA)} - \overset{\frown}{\text{CARA}}$$

$$B = CPRA - CARA - ZCA$$

$$FAPRA = PK + B$$

$$\text{XCENT} = \text{SIN (FAPRA)} \times H$$

$$\text{YCENT} = \text{COS (FAPRA)} \times H$$

When $\dfrac{DB}{2} > H$, then (Figure 180):

$$XX = \left(\frac{DB}{2}\right) \text{SIN (PK)}$$

$$FAPSI = \frac{XX + RF}{H}$$

$$\text{FAPRA} = \text{ARC TAN}\left(\frac{FAPSI}{\sqrt{1 - FAPSI^2}}\right)$$

$$\text{XCENT} = \text{SIN (FAPRA)} \times H$$

$$\text{YCENT} = \text{COS (FAPRA)} \times H$$

Find parabola for evaluating bending stress. Also, find X value and diameter of weakest section. See Figure 181.

$$\text{ALPHA} = 0.1 \qquad \text{(First time only)}$$

$$V = \text{SIN (ALPHA)} \times RF$$

$$VI = \sqrt{RF^2 - V^2}$$

$$T = XCENT - VI$$
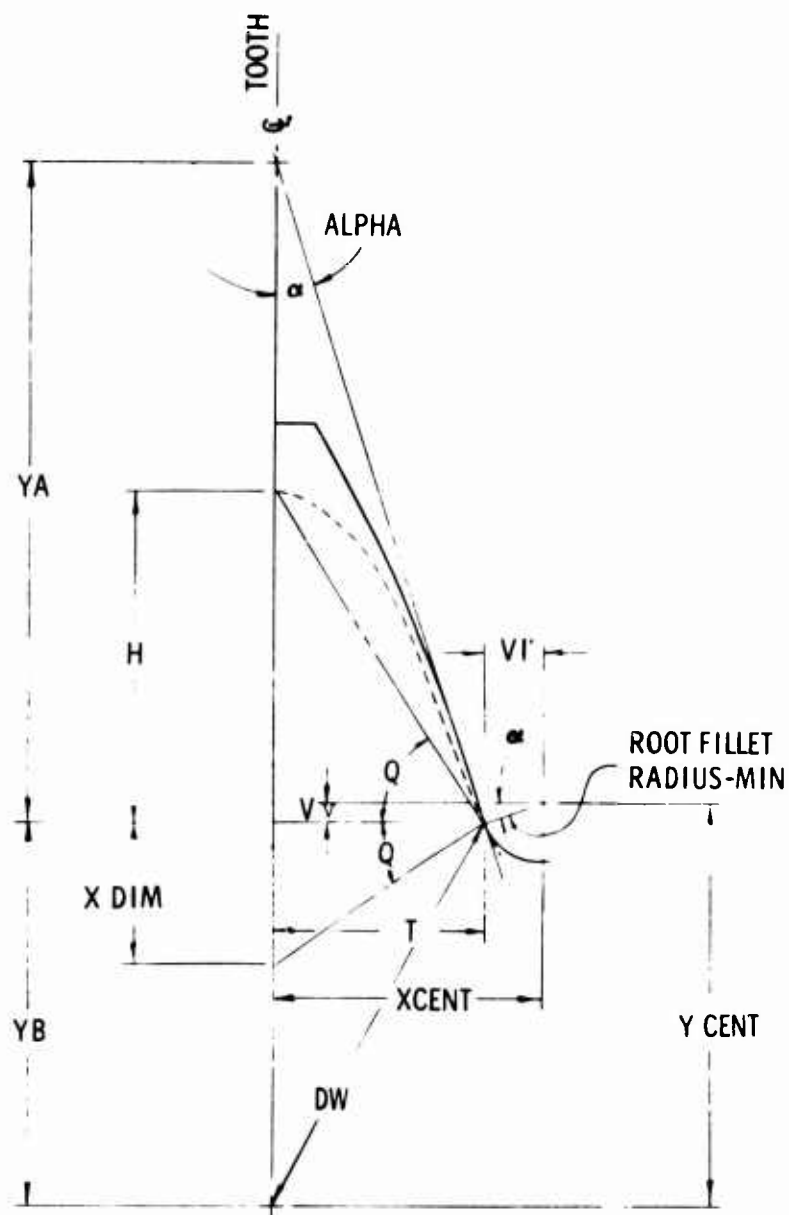
Figure 181.   True Fillet Radius Inscribed Lewis Parabola.

Program

$$YA = \frac{T}{TAN\ (ALPHA)}$$

$$H = (RV - YCENT) + V$$

Reiterate for new value of ALPHA until YA = 2H is satisfied.

$$YB = YCENT - V$$

$$DW = \sqrt{YB^2 + T^2} \times 2$$

$$Q = \text{ARC TAN}\left(\frac{H}{T}\right)$$

$$Q = \frac{\pi}{2} - Q$$

$$\text{XDIM} = T \times \text{TAN}(Q)$$

| AGMA | Program |
|---|---|
| $T = \dfrac{63025 \times Hp}{\eta p}$ | $TQ = \dfrac{63025 \times HORSES}{RPMP}$ |
| $W_t = \dfrac{2 \times T}{\eta p}$ | $WT = \dfrac{2 \times TQ}{RPMP}$ |
| $G = \eta p \times R_{mg}$ | $RPMG = RPMP \times RMG$ |
| $S_h = \rho \dfrac{V^2}{g}$ | $SHOOP = RHO \dfrac{V^2}{386.064}$ |
| $b_1 = b - r_T$ | $B1 = HADD - HTIPR$ |
| $r_1 = \dfrac{b_1^2}{Rp + b_1}$ | $R1 = \dfrac{B1^2}{RP + B1}$ |
| $r_f = r_1 + r_T$ | $RFMI = R1 + HTIPR$ |
| $K_f = 0.22 + \left(\dfrac{T}{r_f}\right)^{0.20}\left(\dfrac{T}{h}\right)^{0.40}$ | $KF = 0.22 + \left(\dfrac{T}{RFMI}\right)^{0.20}\left(\dfrac{T}{H}\right)^{0.40}$ |
| $K_f = 0.18 + \left(\dfrac{T}{r_f}\right)^{0.15}\left(\dfrac{T}{h}\right)^{0.45}$ | $KF = 0.18 + \left(\dfrac{T}{RFMI}\right)^{0.15}\left(\dfrac{T}{H}\right)^{0.45}$ |
| $K_f = 0.14 + \left(\dfrac{T}{r_f}\right)^{0.11}\left(\dfrac{T}{h}\right)^{0.50}$ | $KF = 0.14 + \left(\dfrac{T}{RFMI}\right)^{0.11}\left(\dfrac{T}{H}\right)^{0.50}$ |
| $J = \dfrac{Y}{K_f \times m_\eta}$ | $J = \dfrac{YAGMA}{KF \times MN}$ |
| $S_t = \dfrac{W_t\, K_o}{K_V}\, \dfrac{Pd}{F}\, \dfrac{K_s\, K_m}{J}$ | $SB = \dfrac{WT \times KO}{KV}\, \dfrac{PDX}{FMINP}\, \dfrac{KS \times KM}{J}$ |

Combine bending and centrifugal stress on the modified Goodman diagram. See Figure 182.

From S/N curve in Figure 183, find the life cycle endurance limit.

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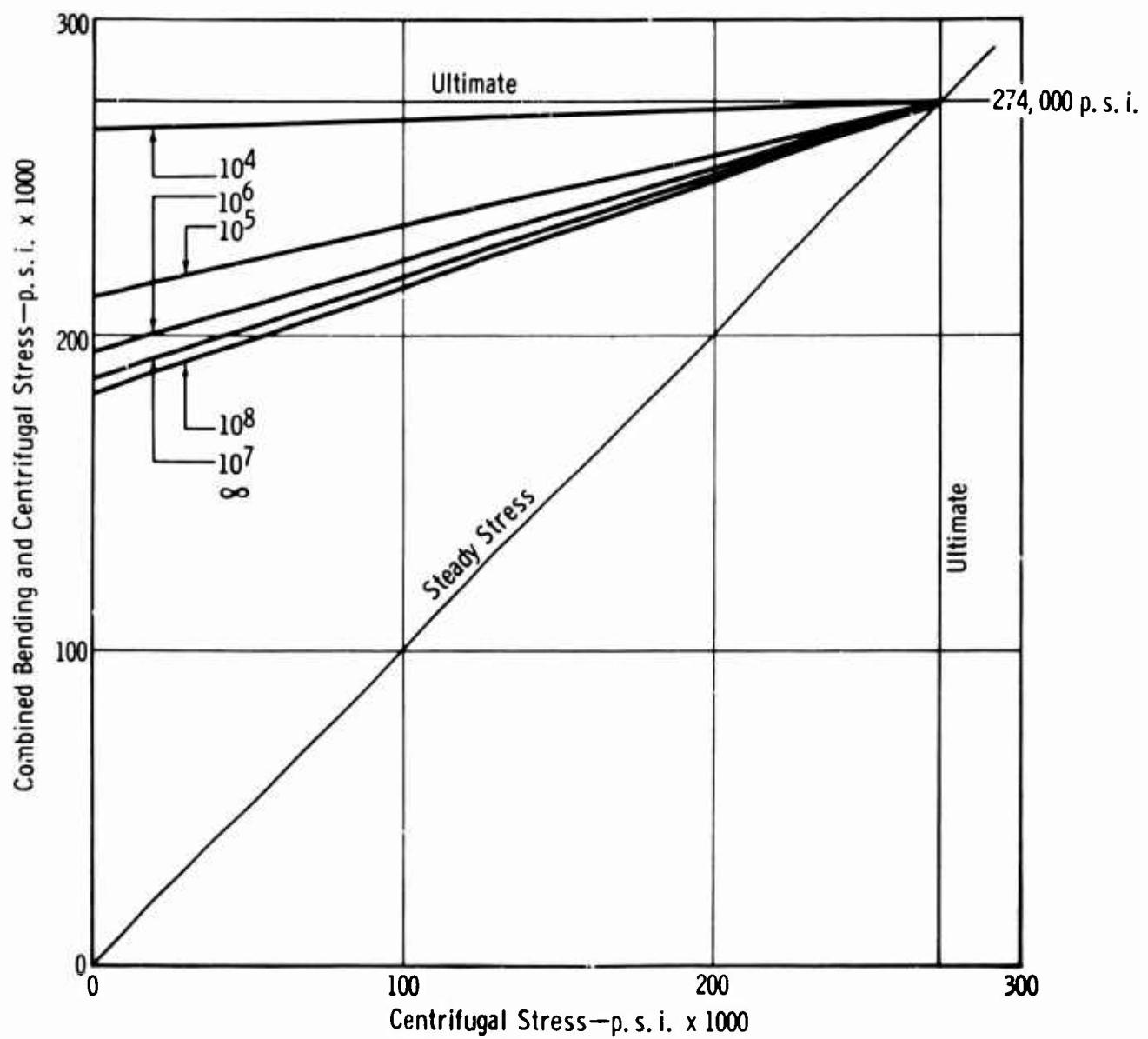
Figure 182.   Modified Goodman Diagram Combining Centrifugal and Bending Stresses.

243
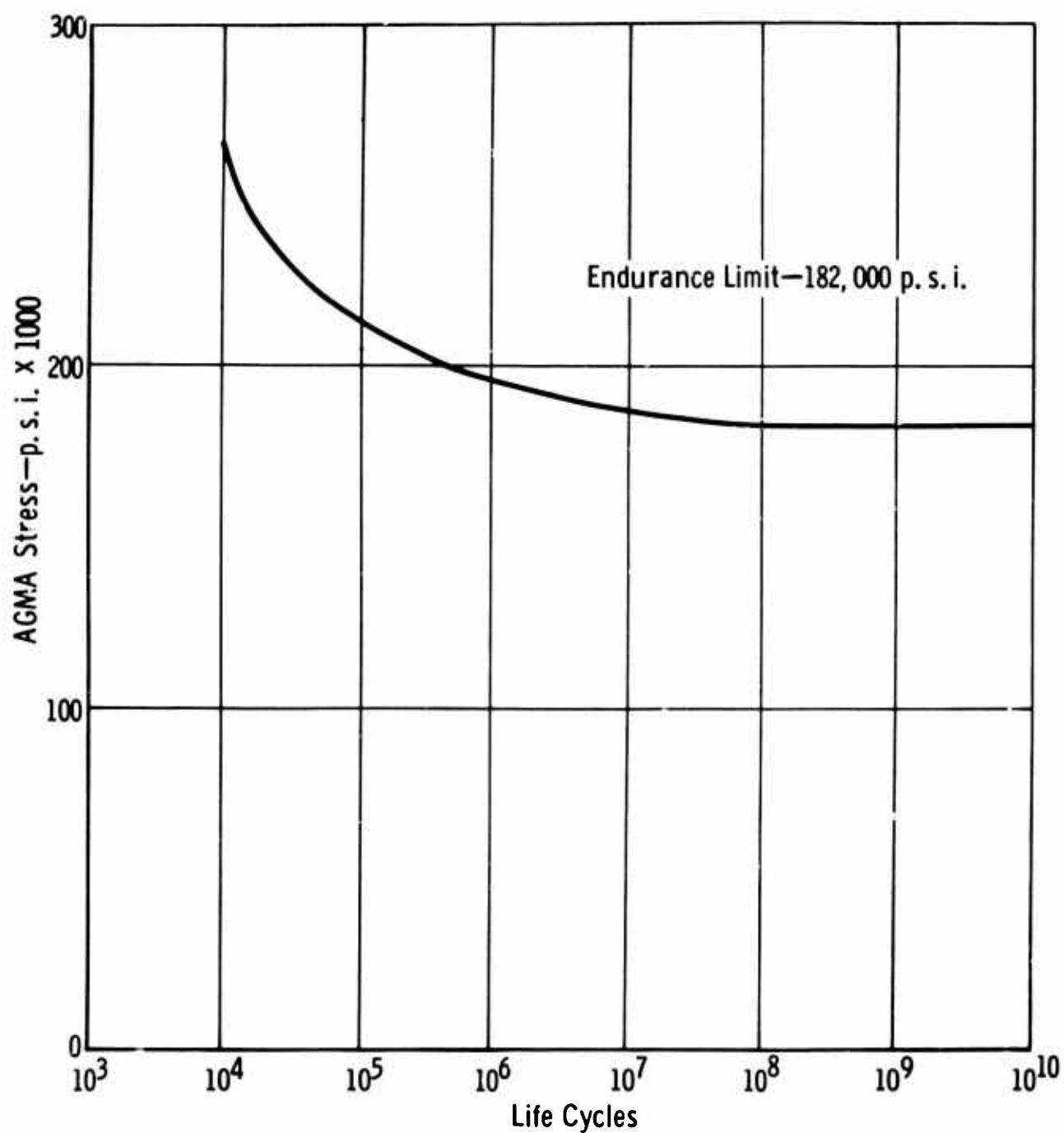
Figure 183. Fatigue Test Gear Endurance Strength for Computer Program.

SOURCE PROGRAM LISTING

The source program is listed on the following pages. Comment cards have been used to define generated symbols within the program. Several subroutines are used and are also listed.

```
C                         *   EXTERNAL SPUR GEARS - FOR  *
C                         *   EVALUATING BENDING STRESS  *
C                         *   PROGRAMED BY M.R. CHAPLIN  *
C                         *   ALLISON,DIV. OF GMC        *
C                         * * * * * * * * * * * * * * *
C
      REAL KT,KR,KM,KO,KV,MN,JP,JG,KFP,KFG,KSP,KSG
      INTEGER CODE                                                        20
      DIMENSION DIAP(6),DIAG(6),FPRA(6),FPDE(6),FPSI(6),FPCO(6),FPTA(6),  30
     *ZFP(6),FGRA(6),FGDE(6),FGSI(6),FGCO(6),FGTA(6),ZFG(6),             40
     *DVP(6),RVP(6),ALPHP(6),TP(6),HP(6),DWP(6),XDIMP(6),               50
     *DVG(6),RVG(6),ALPHG(6),TG(6),HG(6),DWG(6),XDIMG(6),               60
     *SBP(6),SBG(6),SBPHOP(6),SBGHOP(6),YPAGMA(6),YGAGMA(6),            70
     *FILRP(6),FILRG(6),XCYC(5),YPSI (5),            JP(5),JG(5),KFP(5), 80
     *KFG(5),Q1(12),Q3(12)                                               90
      EQUIVALENCE (DIAP(1),DBCP),(DIAP(2),DBSP),(DIAP(3),CP),           100
     *(DIAP(4),DXP),(DIAP(5),DESP),(DIAP(6),DECP),                      110
     *(DIAG(1),DBCG),(DIAG(2),DBSG),(DIAG(3),DG),(DIAG(4),DXG),         120
     *(DIAG(5),DESG),(DIAG(6),DECG)                                     130
C
C  LOGICAL UNIT/MODE (LIN=5   INPUT   5/BCD)
C                    (LOU=6   OUTPUT  6/BCD)
C
      LIN=5                                                              140
      LOU=6                                                              150
    1 READ (LIN,2) ANP,ANG,CNSTD,CUTTER,CODE,HORSES,RPMP,RHO,KT,KR,KM,  160
     *PHIN,PND,BMIN,BMAX,TPMIS,TPMAS,TGMIS,TGMAS,L,                     170
     *DCPMI,DCPMA,DCGMI,DOGMA,FMINP,FMING,BRKP,BRKG,                    180
     *DRPMI,DRPMA,DRGMI,DRGMA,RFMIP,RFMIG,UCP,UCG,KO,KV                 190
    2 FORMAT (2F5.0,F10.0,A6,2X,I2,2F10.0,4F5.0/                        200
     *2F10.0,2F5.0,4F10.0,I2/                                           210
     *6F10.0,2F5.0/                                                     220
     *4F10.0,6F5.0)                                                     230
C
      COMMON RHPA,HPCA,HYP,HRCTRP,           TSA,FCA,YFIL              240
C
      AP=ANP                                                            250
      AG=ANG                                                            260
C
C  DATA STATEMENTS   -   USED TO DEFINE VARIABLE TITLES FOR OUTPUT
C
      DATA (Q1(N),N=1,12) /6HBC (LP,6HC)        ,6HBSTC (,6HLPSTC),    340
     *6HPP (ST,6HD)      ,6HPP (OP,6H)        ,6HESTC (,6HHPSTC),       350
     *6HEC (HP,6HC)      /                                             360
      DATA (Q3(N),N=1,12) /6HBC (HP,6HC)        ,6HBSTC (,6HHPSTC),    370
     *6HPP (ST,6HD)      ,6HPP (OP,6H)        ,6HESTC (,6HLPSTC),       380
     *6HEC (LP,6HC)      /                                             390
C
      DATA SHAPED/6HSHAPED/                                             400
      DATA PINION,GEAR   /6HPINION,6HGEAR   /                          410
C
      DATA (XCYC(M),M=1,5) /4.,5.,6.,7.,8./                            420
      DATA (YPSI (M),M=1,5)/265000.,212000.,198000.,186000.,182000./  430
C
C                         * * * * * * * * * *
C
C
C       RN   -- CONVERT FROM DEGREES TO RADIANS
C       DEGR -- CONVERT FROM RADIANS TO DEGREES
C
      RN=.017453293                                                     450
      DEGR=57.295779513l                                                460
      PI=3.141592653589B                                                470
      IPHI=PHIN                                                         480
```

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```
      FNRA=PHIN*FN                                                           490
      FNSI=SIN(FNRA)                                                         500
      FNCO=COS(FNRA)                                                         510
      FNTA=FNSI/FNCC                                                         520
C
C     PDX            ---DIAMETRAL PITC   (NON STD CENTERS)
C     CSTD           ---STD CENTER DISTANCE
C     FXRA           ---PHI X            (NON STD CENTERS)
C
      CSTD=(ANPGANG)/(2.*PND)                                               530
      IF (CNSTD)             20,19,20                                       540
   19 CNSTD=CSTC                                                            550
   20 PDX=(ANPGANG)/(2.*CNSTD)                                             560
      FX=(CSTC*FNCO)/CNSTD                                                 570
      FXRA=ATAN(SQRT(1.-(FX)**2)/FX)                                       580
      FXSI=SIN(FXRA)                                                        590
      FXCO=COS(FXRA)                                                        600
      FXTA=FXSI/FXCO                                                        610
C
C     ZFN            ---INVOLUTE PHI     (STD CENTERS)
C     ZFX            ---INVOLUTE PHI     (NON STD CENTERS)
C
      IF (CNSTD - CSTD)        604,606,608                                 2260
  604 WRITE (LOU,1000)                                                     2270
      GO TO 21                                                             2280
  606 WRITE (LOU,1001)                                                     2290
      GO TO 21                                                             2300
  608 WRITE (LOU,1002)                                                     2310
   21 WRITE (LOU,1004) AP,NG,CNSTD,CUTTER,CODE,HORSES,RPMP,RHO,KT,KR,KM     270
      IF (L)                  90,92,90                                      280
   90 WRITE (LOU,1005) PHIN,PND,BMIN,BMAX,TPMIS,TPMAS,YGMIS,TGMAS           290
      GO TO 94                                                             300
   92 WRITE (LOU,1006) PHIN,PND,BMIN,BMAX,TPMIS,TPMAS,TGMIS,TGMAS           310
   94 WRITE (LOU,1007) OOPMI,DOPMA,DOGMI,DOGMA,FMINP,FMING,BRKP,BRKG,        320
     *DRPMI,DRPMA,DRGMI,DRGMA,RFMIP,RFMIG,UCP,UCG,KO,KV                      330
      WRITE (LOU,2000)
      ZFN=FNTA-FNRA                                                         620
      ZFX=FXTA-FXRA                                                         630
C
C     AMG            ---GEAR RATIO
C     RMG            ---1/GEAR RATIO
C
      AMG=ANG/ANP                                                           640
      RMG=ANP/ANG                                                           650
C
C             PINION      GEAR
C             DP          CG        - STD PITCH DIA.
C             CBP         CBG       - BASE CIRCLE DIA.
C             DXP         CXG       - NON STD PITCH DIA.
C             OOCBP       COOBG     - OUTSIDE DIA BREAK
C
      DP=ANP/PND                                                            660
      CBP=DP*FNCC                                                           670
      DXP=ANP/PCX                                                           680
      DG=ANG/PND                                                            690
      CBG=DG*FNCC                                                           700
      DXG=ANG/PCX                                                           710
      OODBP=DCPMI-(2.*BRKP)                                                 720
      CODBG=DOGMI-(2.*BRKG)                                                 730
C
C             PINION      GEAR
C             EECP        EECG      - EPSILON   END CONTACT
C             EBCP        EBCG      - EPSILON   BEGIN CONTACT
```

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```
C          EBSP          EBSG      - EPSILON  BEGIN SINGLE TOOTH CONTACT
C          EESP          EESG      - EPSILON  END SINGLE TOOTH CONTACT
C          EOPMA         EOGMA     - EPSILON  OD MAX
C
      EECP=SQRT((DOCBP/CBP)**2-1.)                                    740
      EBCG=SQRT((CODBG/CBG)**2-1.)                                    750
      EBCP=(FXTA*(AMG&1.))-(EBCG*AMG)                                 760
      EECG=(FXTA*(RMG&1.))-(EECP*RMG)                                 770
      EBSP=EECP-((2.*PI)/ANP)                                         780
      EESP=EBCP&((2.*PI)/ANP)                                         790
      EBSG=EECG&((2.*PI)/ANG)                                         800
      EESG=EBCG-((2.*PI)/ANG)                                         810
      EOPMA=SQRT((DCPMA/DBP)**2-1.)                                   820
      EOGMA=SQRT((DOGMA/CBG)**2-1.)                                   830
C
C  DIAMETERS AT ENGAGEMENT CONDITIONS
C          PINION        GEAR
C          DBCP          CBCG      - BEGIN CONTACT
C          DBSP          CBSG      - BEGIN SINGLE TOOTH CONTACT
C          DESP          CESG      - END SINGLE TOOTH CONTACT
C          DECP          CECG      - END CONTACT
C
      DBCP=SQRT((EBCP)**2&1.)*DBP                                     840
      DBSP=SQRT((EBSP)**2&1.)*DBP                                     850
      DESP=SQRT((EESP)**2&1.)*DBP                                     860
      DECP=SQRT((EECP)**2&1.)*DBP                                     870
      DBCG=SQRT((EBCG)**2&1.)*DBG                                     880
      DBSG=SQRT((EBSG)**2&1.)*DBG                                     890
      DESG=SQRT((EESG)**2&1.)*DBG                                     900
      DECG=SQRT((EECG)**2&1.)*DBG                                     910
C
C  AMPMA          -- PROFILE CONTACT RATIO MAX
C  AMPMI          -- PROFILE CONTACT RATIO MIN
C
      AMPMA=((ANG*(ECGMA-FXTA))&(ANP*(EOPMA-FXTA)))/(2.*PI)           920
      AMPMI=((ANG*(EBCG -FXTA))&(ANP*(EECP -FXTA)))/(2.*PI)           930
C
      IF (L)              80,82,80                                    940
C
C  CALCULATE ARC TOOTH THK. FROM CHORDAL THK.
C
   80 AN=(.5*TPMIS)/(.5*DP)                                           950
      AN=ATAN(AN/(SQRT(1.-(AN)**2)))                                  960
      TPMIS=AN*DP                                                     970
      AN=(.5*TPMAS)/(.5*DP)                                           980
      AN=ATAN(AN/(SQRT(1.-(AN)**2)))                                  990
      TPMAS=AN*DP                                                    1000
      AN=(.5*TGMIS)/(.5*DG)                                          1010
      AN=ATAN(AN/(SQRT(1.-(AN)**2)))                                 1020
      TGMIS=AN*DG                                                    1030
      AN=(.5*TGMAS)/(.5*DG)                                          1040
      AN=ATAN(AN/(SQRT(1.-(AN)**2)))                                 1050
      TGMAS=AN*DG                                                    1060
C
C  CALCULATE ARC TOOTH THK. AT THE OPERATING PITCH DIA.   (DXP)
C
   82 TPMIN=DXP*(((TPMIS/DP)&ZFN)-ZFX)                               1070
      TPMAX=DXP*(((TPMAS/DP)&ZFN)-ZFX)                               1080
      TGMIN=DXG*(((TGMIS/DG)&ZFN)-ZFX)                               1090
      TGMAX=DXG*(((TGMAS/DG)&ZFN)-ZFX)                               1100
C
C  CALCULATE PHI AND INVOLUTE PHI AT THE ENGAGEMENT CONDITIONS
C
```

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```
      CALL PHI (CIAP,DEGR,FPRA,FPDE,FPSI,FPCO,FPTA,ZFP,DBP)               1110
      CALL PHI (CIAG,DEGR,FGRA,FGDE,FGSI,FGCO,FGTA,ZFG,DBG)               1120
C
C
C        PINION          GEAR
C        PK              GK        -      ANGLE FROM THE ORGIN OF THE
C                                         INVOLUTE TO THE CENTER LINE OF TOOTH
C        DVP             CVG       -  DIA. TC VERTEX OF PARABOLAS
C        RVP             RVG       -  RAD. TC VERTEX OF PARABOLAS
C
      PK=(TPMIN/CXP)&ZFX                                                  1130
      GK=(TGMIN/CXG)&ZFX                                                  1140
      CO 500 I=1,6                                                        1150
      F=FPTA(I)-FK                                                        1160
      CVP(I)=CBP/COS(F)                                                   1170
      RVP(I)=CVP(I)*.5                                                    1180
      F=FGTA(I)-GK                                                        1190
      DVG(I)=CBG/COS(F)                                                   1200
  500 RVG(I)=CVG(I)*.5                                                    1210
      IF (CUTTER.EQ.SHAPED)   GO TO 512                                   1220
      IF (CCDE - 1)              502,504,506                              1230
  502 CALL HOB (CXP,CRPMI,ANP,PI,RN,TP,HP,DWP,XDIMP,HADDP,HPWP,FILRP,     1240
     *HTIPRP,CVP,6)                                                       1250
      GC TO 508                                                           1260
  504 CALL HOB (CXG,CRGMI,ANG,PI,RN,TG,HG,DWG,XDIMG,HADDG,HPWG,FILRG,     1270
     *HTIPRG,CVG,6)                                                       1280
      GO TO 510                                                           1290
  506 CALL HOB (CXP,CRPMI,ANP,PI,RN,TP,HP,DWP,XDIMP,HADDP,HPWP,FILRP,     1300
     *HTIPRP,CVP,6)                                                       1310
      CALL HOB (CXG,CRGMI,ANG,PI,RN,TG,HG,DWG,XDIMG,HADDG,HPWG,FILRG,     1320
     *HTIPRG,CVG,6)                                                       1330
      GO TO 514                                                           1340
  508 CALL XY (CBG,CRGMI,RFMIG,DEGR,GK,XG,YG)                             1350
      RFMG=RFMIG&UCG                                                      1360
      CALL WEAK (RVG,XG,YG,RFMG,ALPHG,TG,HG,DWG,XDIMG,6)                  1370
      GO TO 514                                                           1380
  510 CALL XY (CBP,CRPMI,RFMIP,DEGR,PK,XP,YP)                             1390
      RFMP=RFMIP&UCP                                                      1400
      CALL WEAK (RVP,XP,YP,RFMP,ALPHP,TP,HP,DWP,XDIMP,6)                  1410
      GO TO 514                                                           1420
  512 CALL XY (CBP,CRPMI,RFMIP,DEGR,PK,XP,YP)                             1430
      CALL XY (CBG,DRGMI,RFMIG,DEGR,GK,XG,YG)                             1440
      RFMP=RFMIP&UCP                                                      1450
      RFMG=RFMIG&UCG                                                      1460
      CALL WEAK (RVP,XP,YP,RFMP,ALPHP,TP,HP,DWP,XDIMP,6)                  1470
      CALL WEAK (RVG,XG,YG,RFMG,ALPHG,TG,HG,DWG,XDIMG,6)                  1480
C
  514 TGP=(63C25.*HORSES)/RPMP                                           1490
      RPMG=RPMP*RMG                                                       1500
      TQG=(63C25.*HORSES)/RPMG                                           1510
      WTP=(2.*TGP)/CXP                                                    1520
      WTG=(2.*TGG)/CXG                                                    1530
      DO 515 I=2,5                                                        1540
$JOB  CHAPLIN,M.           FT4    N84    7893    P57507    002    C10    1
$EXECUTE       IBJOB
$IBJOB N84
$IBFTC N84
C                     * * * * * * * * * * * * * * * *
C                     *   STANDARD AND NON STANDARD  *
      A=FPTA(I)-FK
      BB=COS(A)/FXCG
      BBB=1.5/XDIMP(I)                                                    1560
      BBBB=(SIN(A)/COS(A))/(TP(I)*2.)
      YPAGMA(I)=FGX/(BB*(BBB-BBBB))                                       1580
```

```
      A=FGTA(I)-CK
      BB=COS(A)/FXCO
      BBB=1.5/XCIMG(I)                                                 1600
      BBBB=(SIN(A)/COS(A))/(TG(I)*2.)
  515 YGAGMA(I)=FCX/(BB*(BBB-BBBB))                                    1620
C
      MN=1.0                                                          1630
      IF (CUTTER.EG.SHAPEC)   GO TO 406                                1640
      IF (CCDE - 1)            402,404,400                             1650
  400 B1=HACDP-+TIPRP                                                  1660
      R1=B1**2/((CP*.5)&B1)                                            1670
      RFMIP=R1&+TIPRP                                                  1680
  404 B1=+ACCC-+TIPRG                                                  1690
      R1=B1**2/((CG*.5)&B1)                                            1700
      RFMIG=R1&+TIPRG                                                  1710
      GO TO 406                                                        1720
  402 B1=HACDP-+TIPRP                                                  1730
      R1=B1**2/((CP*.5)&B1)                                            1740
      RFMIP=R1&+TIPRP                                                  1750
  406 IF (IPHI-2C)               408,412,416                           1760
  408 CO 410 I=2,5                                                     1770
      KFP(I)=.22 & ((((TP(I)*2.)/RFMIP)**.20 * ((TP(I)*2.)/HP(I))**.40)  1780
  410 KFG(I)=.22 & ((((TG(I)*2.)/RFMIG)**.20 * ((TG(I)*2.)/HG(I))**.40)  1790
      GO TO 420                                                        1800
  412 CO 414 I=2,5                                                     1810
      KFP(I)=.18 & ((((TP(I)*2.)/RFMIP)**.15 * ((TP(I)*2.)/HP(I))**.45)  1820
  414 KFG(I)=.18 & ((((TG(I)*2.)/RFMIG)**.15 * ((TG(I)*2.)/HG(I))**.45)  1830
      GO TO 420                                                        1840
  416 CO 418 I=2,5                                                     1850
      KFP(I)=.14 & ((((TP(I)*2.)/RFMIP)**.11 * ((TP(I)*2.)/HP(I))**.50)  1860
  418 KFG(I)=.14 & ((((TG(I)*2.)/RFMIG)**.11 * ((TG(I)*2.)/HG(I))**.50)  1870
  420 CO 422 I=2,5                                                     1880
      JP(I)=YPAGMA(I)/(KFP(I)*MN)                                      1890
      JG(I)=YGAGMA(I)/(KFG(I)*MN)                                      1900
      KSP=1.                                                           1902
      KSG=1.                                                           1904
      SBP(I)=((WTP*KO)/KV)*(PDX/FMINP)*((KSP*KM)/JP(I))                1920
  422 SBG(I)=((WTG*KO)/KV)*(PDX/FMING)*((KSG*KM)/JG(I))                1930
      VP=PI*DRPMI*(RPMP/60.)                                           1940
      SHOCPP=RHC*(VP**2/386.064)                                       1950
      VG=PI*DRGMI*(RPMG/60.)                                           1960
      SHOCPG=RHO*(VG**2/386.064)                                       1970
      DO 426 I=2,5                                                     1980
      SBPHOP(I)=SBP(I)&SHOOPP                                          1990
  426 SBGHOP(I)=SBG(I)&SHOOPG                                          2000
C
C  BENDING & HOOF STRESS FROM MODIFIED GOODMAN DIAGRAM
C
      HOOPMA=274COC.                                                   2010
      CO 522 I=2,5                                                     2020
      DIFFP=HCCPMA - SBPHOP(I)                                         2030
      CIFFG=HCOPMA - SBGHOP(I)                                         2040
      EP=HOOPMA-SHOOPP                                                 2050
      EG=HOOPMA-SHOOPG                                                 2060
      AP=(HCOPMA * CIFFP)/EP                                           2070
      AG=(HOOPMA * CIFFC)/EG                                           2080
      SBPHOP(I)=HCOPMA-AP                                              2090
  522 SBGHOP(I)=+COPMA-AG                                              2100
C                                                                      2110
      IF (SBPHOP(5) - 274COO.)         526,526,524                     2120
  524 WRITE (LOU,1003) PINION                                          2130
  526 IF (SBGHOP(2) - 274000.)         530,530,528                     2150
  528 WRITE (LOU,1003) GEAR                                            2160
```

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  530 IF (SBPHOP(5)-182C00.) 624,624,626                                    2180
  624 WRITE (LOU,10C8) PINION                                               2190
      GO TO 628                                                            2200
  626 CALL CISCOT (SBPHCP(5),DUMA,YPSI,XCYC,DUMB,-31,5,0,EXP)               2210
  628 IF (SBGHOP(2)-182C0C.) 63C,630,632                                    2220
  630 WRITE (LOU,1008) GEAR                                                 2230
      GO TO 609                                                            2240
  632 CALL CISCCT (SBGHCP(2),DUMA,YPSI,XCYC,DUMB,-31,5,0,EXG)               2250
C
  609 WRITE (LOU,1CC9) PINION,GEAR                                          2320
      N=3                                                                  2330
      CC 201 I=2,5                                                         2340
      WRITE (LOU,1010) C1(N),Q1(N&1),SBPHOP(I),Q3(N),Q3(N&1),SBGHOP(I)     2350
  201 N=N&2                                                                2360
      WRITE (LCU,999) PINION,GEAR
  999 FCRMAT (///26X36HB E N D I N G   S T R E S S   (AGMA)//21X,A6,28X
     *,A6)
      N=3
      CO 202 I=2,5
      WRITE (LOU,1010) C1(N),Q1(N&1),SBP(I),Q3(N),Q3(N&1),SBG(I)
  202 N=N&2
      WRITE (LOU,9999) SHOOPP,SHOOPG
 9999 FORMAT (///20X11HHOOP STRESS/10X6HPINION23X4HGEAR/F19.4,15XF12.4)
      IF (SBPHOP(5)-182C00.) 612,612,610                                   2370
  610 WRITE (LOU,1011) PINION,EXP                                          2380
  612 IF (SBGHOP(2)-182C0C.) 1,1,614                                       2390
  614 WRITE (LOU,1011) GEAR,EXG                                            2400
 1000 FORMAT(1H134X23HNCN STANDARD SPUR GEARS/35X25HDECREASED CENTER DIS
     *TANCE)
 1001 FORMAT(1H134X19HSTANDARD SPUR GEARS/35X24HSTANDARD CENTER DISTANCE
     *)
 1002 FORMAT(1H134X23HNCN STANDARD SPUR GEARS/35X25HINCREASED CENTER CIS
     *TANCE)
 1003 FORMAT(///5X15HBENDING STRESS A6, 9H AT HPSTC/4X31HEXCEEDS ULTIMAT
     *E OF 27400C. PSI)
 1004 FORMAT (///35X35HI N P U T   D A T A   S E C T I C N///5X
     *15HNUMBER CF TEETH9X6HCENTER9X          1H*7X4HCODE7X2HHP11X3HRPM5X
     *7HDENSITY6X2HKT7X2HKR7X2HKM/5X15HPINION      GEAR8X
     *8HDISTANCE29X          18HPINION   LB/CU. IN/5X14,6X14,F17.6,8XA6,
     *2X12,1X2F14.4,4F9.4)
 1005 FORMAT (/5X8HPRESSURE5X9HDIAMETRAL9X8HBACKLASH8X25HCHORDAL TOOTH T
     *HK -PINION6X23HCHORDAL TOOTH THK -GEAR/5X5HANGLE8X5HPITCH11X3HMIN
     *6X3HMAX6X25HMIN    (STD PD)   MAX6X24HMIN    (STD PD)    MAX
     */F14.6,F13.6,F11.4,F9.4,F13.6,3X2F14.6,F15.6)
 1006 FORMAT (/5X8HPRESSURE5X9HDIAMETRAL9X8HBACKLASH8X21HARC TOOTH THK -
     *PINION10X19HARC TOOTH THK -GEAR/5X5HANGLE8X5HPITCH11X3HMIN6X3HMAX
     *6X21HMIN    (STD PD)    MAX1CX20HMIN    (STD PD)  MAX /F14.6,F13.6,
     *F11.4,F9.4,2F13.6,7X2F11.6)
 1007 FORMAT (///5X2CHOUTSIDE DIA - PINION9X2CHOUTSIDE DIA  -  GEAR7X
     *18HFACE WICTH  -  MIN4X13HMAX TIP BREAK/5X3HMIN14X3HMAX9X3HMIN14X
     *3HMAX7X6HPINION8X4HGEAR4X6HPINION3X4HGEAR/3X2F11.6,7X2F11.6,5X
     *2F10.6,3X2F7.4//5X20HROOT DIA    - PINICN9X20HROOT CIA    -  GEAR
     *7X18HFILLET RADIUS -MIN4X13HMAX  UNDERCUT5X2HKO7X2HKV/5X3HMIN14X
     *3HMAX9X3HMIN14X3HMAX7X6HPINION8X4HGEAR4X6HPINION3X4HGEAR/3X
     *2F11.6,7X2F11.6,1X2F12.6,3X2F7.4,2F9.4)
 1008 FORMAT (///5X15HBENDING STRESS-A6,17H-AT HPSTC IS LESS/4X
     *56HTHAN THE ENDURANCE LIMIT OF 182000. PSI - INFINITE LIFE.)
 1009 FORMAT(///26X27HB E N D I N G    S T R E S S3X10H(CCMBINED)//
     *21X,A6,28X,A6)
 1010 FCRMAT (10X,2A6,F15.4,5X,2A6,F15.4)
 1011 FORMAT (//5X12HLIFE CYCLES ,A6,19H 10 TO A EXPONET CF,F7.2)
 2000 FORMAT (1H134X37HO U T P U T   D A T A   S E C T I C N)
      GO TO 1
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      END
$IBFTC PHI.
C   SUBROUTINE PHI -CALC. PRESSURE ANGLES AND INVOLUTE ANGLES AT
C   ENGAGEMENT CONDITIONS
C
      SUBROUTINE PHI (DIA,DEGR,FRA,FDE,FSI,FCC,FTA,ZF,DB)              PHI     1
      DIMENSION DIA(6),FRA(6),FDE(6),FSI(6),FCO(6),FTA(6),ZF(6)        PHI     2
      DO 10 I=1,6                                                      PHI     3
      F=DB/DIA(I)                                                      PHI     4
      FRA(I)=ATAN(SQRT(1.-(F)**2)/F)                                   PHI     5
      FDE(I)=FRA(I)*DEGR                                               PHI     6
      FSI(I)=SIN(FRA(I))                                               PHI     7
      FCO(I)=COS(FRA(I))                                               PHI     8
      FTA(I)=FSI(I)/FCO(I)                                             PHI     9
      ZF(I)=FTA(I)-FRA(I)                                             PHI    10
   10 CONTINUE                                                        PHI    11
      RETURN                                                          PHI    12
      END                                                             PHI    13
$IBFTC XY.
C   SUBROUTINE XY --CALCULATES COORDINATES TO CENTER OF FILLET RADIUS  XY      1
C                                                                     XY      2
      SUBROUTINE XY (DB,DR,RF,DEGR,PK,          X,Y)                   XY      3
      H=(DR/2.)&RF                                                     XY      4
      IF ((DB/2.)-H)          10,1C,12                                 XY      5
   10 CPR=(DB/2.)/H                                                    XY      6
      CPRA=ATAN(SQRT(1.-(CPR)**2)/CPR)                                 XY      7
      OPP=SQRT((H)**2-(DB/2.)**2)                                      XY      8
      A=CPP-RF                                                         XY      9
      H1=SQRT((A)**2&(DB/2.)**2)                                       XY     10
      CA=(DB/2.)/H1                                                    XY     11
      CARA=ATAN(SQRT(1.-(CA)**2)/CA)                                   XY     12
      ZCA=(SIN(CARA)/COS(CARA))-CARA                                   XY     13
      B=(CPRA-CARA)-ZCA                                                XY     14
      FAPRA=PK&B                                                       XY     15
   11 X=SIN(FAPRA)*H                                                   XY     17
      Y=COS(FAPRA)*H                                                   XY     18
      RETURN                                                          XY     19
   12 XX=(DB/2.)*SIN(PK)                                               XY     20
      FAPSI=(XX&RF)/H                                                  XY     21
      FAPRA=ATAN(FAPSI/(SQRT(1.-(FAPSI'**2))))                         XY     22
      GO TO 11                                                         XY     24
      END                                                             XY     25
$IBFTC WEAK.
C   SUBROUTINE WEAK CALCS. THE DIA. OF THE WEAKEST SECTION (DW) BY
C   INSCRIBING THE LARGEST PARABOLA THAT WILL FIT THE GEAR TOOTH SHAPE.
C
      SUBROUTINE WEAK (RV,XCENT,YCENT,RF,ALPHA,T,H,DW,XDIM,NOD)       WEAK    1
      DIMENSION RV(6),ALPHA(6),T(6),H(6),XDIM(6),DW(6)                WEAK    2
      DO 10 I=1,NOD                                                    WEAK    3
      ALPHA(I)=.1                                                      WEAK    4
      DELTA=.1                                                         WEAK    5
  144 V=SIN(ALPHA(I))*RF                                               WEAK    6
      V1=SQRT((RF)**2-(V)**2)                                          WEAK    7
C      T --  HALF CHORD AT THE WEAKEST SECTION                        WEAK    8
      T(I)=XCENT-V1                                                    WEAK    9
      YA=T(I)/(SIN(ALPHA(I))/COS(ALPHA(I)))                            WEAK   10
C      H --  TOOTH HEIGHT FROM WEAKEST SECTION TO VERTEX OF PARABOLA  WEAK   11
      H(I)=(RV(I)-YCENT)&V                                             WEAK   12
      YAP=YA*.5                                                        WEAK   13
      IF (YAP - H(I))          146,150,148                            WEAK   14
  146 ALPHA(I)=ALPHA(I)-DELTA                                         WEAK   15
      DELTA=.1*DELTA                                                   WEAK   16
      IF (.000000C0i-DELTA)    144,150,150                            WEAK   17
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  148 ALPHA(I)=ALPHA(I)&DELTA                                          WEAK   18
      GO TO 144                                                        WEAK   19
  150 YB=YCENT-V                                                       WEAK   20
C        DW  --   WEAKEST SECTION DIAMETER                             WEAK   21
      DW(I)=SQRT((YB)**2&(T(I))**2)*2.                                 WEAK   22
      Q=ATAN(H(I)/T(I))                                                WEAK   23
      Q=1.5707963-Q                                                    WEAK   24
C        XDIM-      X DIMENSION                                        WEAK   25
      XDIM(I)=T(I)*(SIN(Q)/COS(Q))                                     WEAK   26
   10 CONTINUE                                                         WEAK   27
      RETURN                                                           WEAK   28
      END                                                              WEAK   29
$IBFTC HOB.
C  SUBROUTINE HOB  --
C
C
      SUBROUTINE HOB (DX,DRPM,ANP,PI,RN,T,H,DW,XDIM,HADD,HPW,FILR,      HOB    1
     *HTIPR,DVP,ADD)
      DIMENSION DVP(6),XDIM(6),DW(6),T(6),H(6),P1(6),YYFIL(6),DIAP(6),  HOB    2
     *DIAG(6),FILR(6)                                                   HOB    3
      EQUIVALENCE (DIAP(1),DBCP),(DIAP(2),DBSP),(DIAP(3),CP),(DIAP(4),         70
     *DXP),(DIAP(5),DESP),(DIAP(6),DECP),(DIAG(1),DBCG),(DIAG(2),              80
     *DBSG),(DIAG(3),DG),(DIAG(4),DXG),(DIAG(5),DESG),(DIAG(6),DECG)          90
C  LOGICAL UNIT/MODE   (LIN=5   INPUT 5/BCD)
C                      (LOU=6   OUTPUT 6/BCD)
C
      LIN=5                                                            HOB    4
      LOU=6                                                            HOB    5
      READ (LIN,2) HTT,HADD,HLEAD,HPA,HTIPR,HPW                        HOB    6
    2 FORMAT (6F10.0)                                                  HOB    7
      COMMON RHPA,HPCA,HYP,HRCTRP,        TSA,FCA,YFIL                 HOB    8
      WRITE (LOU,1008) HTT,HADD,HLEAD,HPA,HTIPR,HPW                    HOB    9
 1008 FORMAT (//3X9HHOB  DATA//5X21HTOOTH THK.   ADDENDUM7X4HLEAD4X     HOB
     *24HPRESSURE ANGLE   TIP RAD.7X3HHPW/6F13.6)                      HOB
      REAL INC                                                         HOB    7
      TSA=PI/ANP                                                       HOB    8
      DHPA=(ANP*HLEAD)/PI                                              HOB    9
      HADDN=.5*(DHPA-DRPM)                                             HOB   10
      HPAR=HPA*RN                                                      HOB   11
      HTTN=HTT&2.*(HADDN-HADD)*TAN(HPAR)                               HOB   12
      HTTR=.5*HTT-HADD*TAN(HPAR)                                       HOB   13
      HA=HTTR-(HTIPR-HPW)/COS(HPAR)                                    HOB   14
      HRCTRX=HA&HTIPR*TAN(HPAR)                                        HOB   15
      RHPA=.5*DHPA                                                     HOB   16
      HRCTRP=HADD-HTIPR                                                HOB   17
      HPCA=ATAN(HRCTRX/HRCTRP)                                         HOB   18
      HYP=HRCTRP/COS(HPCA)                                             HOB   19
C  FIND PARABOLA TANGENT TO GENERATED FILLET                          HOB   20
      FTCA=(PI/2.)-TSA                                                 HOB   21
      DO 25 I=1,NDD                                                    HOB   22
      INC=.1*HTTN                                                      HOB   23
      PPPOS=0.                                                         HOB   24
    5 PPPOS=PPPOS&INC                                                  HOB   25
      CALL GENFIL (PPPOS, T(I),YYFIL(I),DW(I),FCA,HTIPR)              HOB   26
      FTPA=PI-(FTCA&FCA)                                               HOB   27
      FPARA=(PI/2.)-FTPA                                               HOB   28
      AB= T(I)/TAN(FPARA)                                              HOB   29
      H(I)=.5*DVP(I)-YYFIL(I)                                          HOB   30
      K=1000000.*(AB-2.*H(I))                                         HOB   31
      IF (K)              5,15,10                                      HOB   32
   10 PPPOS=PPPOS-INC                                                  HOB   33
      INC=.1*INC                                                       HOB   34
      GO TO 5                                                          HOB   35
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   15 PI(I)= T(I)**2/(2.*H(I))                                         HOB   36
C FINC RADIUS OF CURVATURE OF GENERATED FILLET AT TANGENT OF PARABOLA   HOB   37
      SIDEA=YFILGHADON-RHPA                                             HOB   38
      HYPA=SICEA/COS(FCA)                                               HOB   39
      ANGLEA=.5*((PI/2.)GFCA)                                           HOB   40
      FILR(I)=HYFA*TAN(ANGLEA)                                          HOB   41
C FIND  --X-- VALUE FOR PARABOLA                                        HOB   42
      ANGLED=ATAN( T(I)/H(I))                                           HOB   43
      ADJ= T(I)/SIN(ANGLED)                                             HOB   44
   25 XCIM(I)=ADJ/COS(ANGLED)-H(I)                                      HOB   45
      RETURN                                                            HOB   46
      END                                                               HOB   47
$IBFTC GENFI.
C   SUBROUTINE GENFIL  -
C
C
      SUBROUTINE GENFIL (PPPOS,XIFIL,YTFIL,DFIL,FCA,HTIPR)           GENFIL  1
      CIMENSION DIAP(6),DIAG(6)
      ECUIVALENCE (DIAP(1),DBCP),(DIAP(2),DBSP),(DIAP(3),CP),(DIAP(4),     70
     *DXP),(DIAP(5),DESP),(DIAP(6),DECP),(DIAG(1),DBCG),(CIAG(2),          80
     *CBSG),(CIAC(3),DG),(DIAG(4),DXG),(DIAG(5),DESG),(DIAG(6),DECG)       90
      COMMON RHPA,HPCA,HYP,HRCTRP,        TSA,FCA,YFIL                HOB    8
      PPA=PPPCS/RHPA                                                 GENFIL  3
      PHA=ATAN(PFA)                                                  GENFIL  4
      HPCTR=RHPA/COS(PHA)                                            GENFIL  5
      PPPHA=PPA-FHA                                                  GENFIL  6
      HPCX=HPCTR*SIN(PPPHA)                                          GENFIL  7
      HPCY=HPCTR*CGS(PPPHA)                                          GENFIL  8
      RCTRA=HPCA&PPA                                                 GENFIL  9
      RCTX=HYP*SIN(RCTRA)-HPCX                                       GENFIL10
      RCTY=HPCY-HYP*COS(RCTRA)                                       GENFIL11
      IF (PPPOS)        1C,10,5                                      GENFIL
   10 XFIL=RCTX                                                      GENFIL
      YFIL=RCTY-HTIPR                                                GENFIL
      GO TO 15                                                       GENFIL
    5 FCPLA=ATAN(HRCTRP/PPPOS)                                       GENFIL12
      FCA=FCPLA-FPA                                                  GENFIL13
      XFIL=RCTXG+TIPR*CCS(FCA)                                       GENFIL14
      YFIL=RCTY-HTIPR*SIN(FCA)                                       GENFIL15
   15 FSA=ATAN(XFIL/YFIL)                                            GENFIL16
      FTA=TSA-FSA                                                    GENFIL17
      RFIL=YFIL/COS(FSA)                                             GENFIL18
      XTFIL=RFIL*SIN(FTA)                                            GENFIL19
      YTFIL=RFIL*COS(FTA)                                            GENFIL20
      CFIL=2.0*RFIL                                                  GENFIL21
      RETURN                                                         GENFIL22
      END                                                            GENFIL23
$IBFTC DISCOD  LIST
CDISCCT                                                               DISCO0010
      SUBROUTINE CISCOT (XA,ZA,TABX,TABY,TABZ,NC,NY,NZ,ANS)          DISCO0020
      CIMENSION TABX(50C),TABY(500),TABZ(500),NPX(8),NPY(8),YY(8)    DISCO0030
      CALL UNS (NC,IA,ICX,IDZ,IMS)                                   DISCO0050
      IF (NZ-1) 5,5,10                                               DISCO0060
    5 CALL CISSER (XA,TABX,1,NY,IDX,NN)                              DISCO0070
      NNN=ICXG1                                                      DISCO0080
      CALL LAGRAN (XA,TABX(NN),TABY(NN),NNN,ANS)                     DISCO0090
      GU TO 7C                                                       DISCO0100
   10 ZARG=ZA                                                        DISCO0110
      IP1X=IDXG1                                                     DISCO0120
      IP1Z=IDZG1                                                     DISCO0130
      IF (IA)    15,25,15                                            DISCO0140
   15 IF (ZARG-TABZ(NZ))    25,25,2C                                 DISCO0150
   20 ZARG=TABZ(NZ)                                                  DISCO0160
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    25 CALL CISSER (ZARG,TABZ,1,NZ,IOZ,NPZ)                       DISCO170
       NX=NY/NZ                                                   DISCO180
       NPZL=NPZ&ICZ                                               DISCO190
       I=1                                                        DISCO200
       IF (IMS)     3C,30,4C                                      DISCO210
    30 CALL CISSER (XA,TABX,1,NX,IDX,NPX)                         DISCO220
       DO 35  JJ=NPZ,NPZL                                         DISCO230
       NPY(I)=(JJ-1)*NX&NPX(1)                                    DISCO240
       NPX(I)=NPX(1)                                              DISCO250
    35 I=I&1                                                      DISCO260
       GO TO 50                                                   DISCO270
    40 CO 45 JJ=NFZ,NPZL                                          DISCO280
       IS=(JJ-1)*NX&1                                             DISCO290
       CALL CISSER (XA,TABX,IS,NX,IDX,NPX(I))                     DISCO300
       NPY(I)=NPX(I)                                              DISCO310
    45 I=I&1                                                      DISCO320
    50 CO 55  I=1,IP1Z                                            DISCO330
       NLOC=NPX(I)                                                DISCO340
       NLOCY=NPY(I)                                               DISCO350
    55 CALL LAGRAN (XA,TABX(NLOC),TABY(NLOCY),IP1X,YY(I))         DISCO360
       CALL LAGRAN (ZARG,TABZ(NPZ),YY,IP1Z,ANS)                   DISCO370
    70 RETURN                                                     DISCO380
       END                                                        DISCO390
$IBFTC LAGRAD  LIST
CLAGRAN                                                           LAGRO010
       SUBROUTINE LAGRAN (XA,X,Y,N,ANS)                           LAGRO020
       DIMENSION X(200),Y(200)                                    LAGRO030
       SUM=0.0                                                    LAGRO050
       DO 3   I=1,N                                               LAGRO060
       PROD=Y(I)                                                  LAGRO070
       CO 2 J=1,N                                                 LAGRO080
       A=X(I)-X(J)                                                LAGRO090
       IF (A)   1,2,1                                             LAGRO100
     1 B=(XA-X(J))/A                                              LAGRO110
       PROD=PROD*E                                                LAGRO120
     2 CONTINUE                                                   LAGRO130
     3 SUM=SUM&PROD                                               LAGRO140
       ANS=SUM                                                    LAGRO150
       RETURN                                                     LAGRO160
       END                                                        LAGRO170
$IBFTC UNSD    LIST
CUNS                                                              UNS 0010
       SUBROUTINE UNS (IC,IA,IDX,IOZ,IMS)                         UNS 0020
       IF (IC)    5,5,10                                          UNS 0030
     5 IMS=1                                                      UNS 0040
       NC=-IC                                                     UNS 0050
       GO TO 15                                                   UNS 0060
    10 IMS=0                                                      UNS 0070
       NC=IC                                                      UNS 0080
    15 IF (NC-100)   20,25,25                                     UNS 0090
    20 IA=0                                                       UNS 0100
       GC TO 30                                                   UNS 0110
    25 IA=1                                                       UNS 0120
       NC=NC-100                                                  UNS 0130
    30 IDX=NC/10                                                  UNS 0140
       IOZ=NC-IDX*10                                              UNS 0150
       RETURN                                                     UNS 0160
       END                                                        UNS 0170
$IBFTC CISSED  LIST
CCISSER                                                           DISS0010
       SUBROUTINE CISSER (XA,TAB,I,NX,IO,NPX)                     DISS0020
       DIMENSION TAB(2000)                                        DISS0030
       NPT=IC&1                                                   DISS0050
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      NPB=NPT/2                                                    DISS0060
      NPU=NPT-NPB                                                  DISS0070
      IF (NX-NPT)    10,5,10                                       DISS0080
    5 NPX=1                                                        DISS0090
      RETURN                                                       DISS0100
   10 NLCW=I&NPB                                                   DISS0110
      NUPP=I&NX-(NPU&1)                                            DISS0120
      CO 15 II=NLCW,NUPP                                           DISS0130
      NLOC=II                                                      DISS0140
      IF (TAB(II)-XA)    15,20,20                                  DISS0150
   15 CONTINUE                                                     DISS0160
      NPX=NUPP-NFB&1                                               DISS0170
      RETURN                                                       DISS0180
   20 NL=NLOC-NPB                                                  DISS0190
      NU=NL&IC                                                     DISS0200
      CC 25  JJ=NL,NU                                              DISS0210
      NCIS=JJ                                                      DISS0220
      IF (TAB(JJ)-TAB(JJ&1))    25,30,25                           DISS0230
   25 CONTINUE                                                     DISS0240
      NPX=NL                                                       DISS0250
      RETURN                                                       DISS0260
   30 IF (TAB(NCIS)-XA)     40,35,35                               DISS0270
   35 NPX=NCIS-IC                                                  DISS0280
      RETURN                                                       DISS0290
   40 NPX=NCIS&1                                                   DISS0300
      RETURN                                                       DISS0310
      END                                                          DISS0320
$CATA
32.0 100.0           SHAPED    3755.C     13820.C    .283 1.1451.0  1.0
25.0       6.0       .012 .018 .2778      .2808      .2278      .2308       0
5.702      5.707     16.929   16.934      2.545      2.490      .020 .020
4.954      4.974     16.1800  16.2000     .030 .030 .0    .0    1.0  1.450
```

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STD & NON STD EXTERNAL SPUR GEARS
BENDING STRESS

☒ SHAPED
☒ HOBBED

IBM 7094 DATA SHEET

| NO. OF TEETH PINION GEAR | NON STD CENTER DISTANCE | * | HORSEPOWER | RPM PINION | DENSITY LB/IN³ | $K_T$ | $K_R$ | $K_m$ | IDENTIFICATION NUMBER |
|---|---|---|---|---|---|---|---|---|---|
| 32.0 100.0 | ☒ | | 3755.5 | 13820.0 | .283 | 1.1451 | 1.0 | 1.0 | 1 |

| PRESSURE ANGLE | DIAMETRAL PITCH | BACKLASH MIN MAX | ARC OR CHORDAL TOOTH THK PINION (STD Dp)-MIN | (STD Dp)-MAX | ARC OR CHORDAL TOOTH THK GEAR (STD DA)-MIN | (STD DA)-MAX | |
|---|---|---|---|---|---|---|---|
| 25.0 | 6.0 | .012 .018 | .2778 | .2808 | .2278 | .2308 | 2 |

| OUTSIDE DIAMETER-PINION MIN MAX | OUTSIDE DIAMETER-GEAR MIN MAX | FACE WIDTH-MIN PINION GEAR | MAX TIP BREAK PINION GEAR | |
|---|---|---|---|---|
| 5.702 5.707 | 16.929 16.934 | 2.545 2.490 | .020 .020 | 3 |

| ROOT DIAMETER-PINION MIN MAX | ROOT DIAMETER-GEAR MIN MAX | FILLET RAD-MIN PINION GEAR | MAX UNDERCUT PINION GEAR | $K_o$ | $K_y$ | |
|---|---|---|---|---|---|---|
| 4.954 4.974 | 16.180 16.934 | .030 .030 | .0 .0 | 1.0 | 1.00 | 4 |

| TOOTH THK (HTT) | ADDENDUM (HADD) | LEAD (HLEAD) | PRESSURE ANGLE (HPA) | TIP RADIUS (HTIPR) | HPW | SEE FIGURE 170 PINION | |
|---|---|---|---|---|---|---|---|
| | | | | | | | 5 |

| TOOTH THK (HTT) | ADDENDUM (HADD) | LEAD (IN LEAD) | PRESSURE ANGLE (HPA) | TIP RADIUS (HTIPR) | HPW | SEE FIGURE 170 GEAR | |
|---|---|---|---|---|---|---|---|
| | | | | | | | 6 |

EXAMPLE

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INPUT DATA SHEET

### I N P U T   D A T A   S E C T I O N

| NUMBER OF TEETH | | NON STD CENTER | * | CODE | HP | RPM | DENSITY |
| PINION | GEAR | DISTANCE | | | | PINION | LB/CU. IN |
| 32 | 100 | 11.000000 | SHAPED | -C | 3755.0000 | 13870.0000 | 0.2330 |

| PRESSURE | DIAMETRAL | BACKLASH | | ARC TOOTH THK -PINION | | | ARC TOOTH |
| ANGLE | PITCH | MIN | MAX | MIN | (STD PD) | MAX | MIN | (STD |
| 25.000000 | 6.000000 | 0.0120 | 0.0180 | 0.277800 | 0.280800 | | 0.227800 |

| OUTSIDE DIA - PINION | | OUTSIDE DIA - GEAR | | FACE WIDTH - MIN | | MAX TIP R? |
| MIN | MAX | MIN | MAX | PINION | GEAR | PINION  G |
| 5.702000 | 5.707000 | 16.929000 | 16.934000 | 2.545000 | 2.490000 | 0.0220 C.0 |

| ROOT DIA - PINION | | ROOT DIA - GEAR | | FILLET RADIUS -MIN | | MAX UNDER |
| MIN | MAX | MIN | MAX | PINION | GEAR | PINION  G |
| 4.954000 | 4.974000 | 16.180000 | 16.200000 | 0.030000 | 0.030000 | 0.0000 0.0 |

BENDING STRESS-PINION-AT HPSTC IS LESS
THAN THE ENDURANCE LIMIT OF 182000. PSI - INFINATE LIFE.

BENDING STRESS-GEAR -AT HPSTC IS LESS
THAN THE ENDURANCE LIMIT OF 182000. PSI - INFINATE LIFE.

### B E N D I N G   S T R E S S

| PINION | | GEAR | |
| BSTC (LPSTC) | 32696.3555 | BSTC (HPSTC) | 125957.7188 |
| PP (STD) | 68065.4707 | PP (STD) | 94310.6230 |
| PP (OP) | 68065.4707 | PP (OP) | 94310.6230 |
| ESTC (HPSTC) | 123804.7285 | ESTC (LPSTC) | 24721.3926 |

### B E N D I N G   S T R E S S   (AGMA)

| PINION | | GEAR | |
| BSTC (LPSTC) | 31572.2649 | BSTC (HPSTC) | 121227.6045 |
| PP (STD) | 65725.4014 | PP (STD) | 90768.9580 |
| PP (OP) | 65725.4014 | PP (OP) | 90768.9580 |
| ESTC (HPSTC) | 119548.3613 | ESTC (LPSTC) | 23793.0264 |

### HOOP STRESS

| PINION | GEAR |
| 9420.0336 | 10289.5746 |

**OUTPUT SHEET**

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### I N P U T   D A T A   S E C T I O N

| ON STD CENTER DISTANCE | * | CODE | HP | RPM PINION | DENSITY LB/CU. IN | KT | KR | KM |
|---|---|---|---|---|---|---|---|---|
| 11.000000 | SHAPED | -C | 3755.0000 | 13820.0000 | 0.2330 | 1.1450 | 1.0000 | 1.0000 |

| BACKLASH | | ARC TOOTH THK -PINION | | | ARC TOOTH THK -GEAR | | |
|---|---|---|---|---|---|---|---|
| MIN | MAX | MIN | (STD PD) | MAX | MIN | (STD PD) | MAX |
| 0.0120 | 0.0180 | 0.277800 | | 0.280800 | 0.227800 | | 0.2 2800 |

| OUTSIDE DIA - GEAR | | FACE WIDTH - MIN | | MAX TIP BREAK | |
|---|---|---|---|---|---|
| MIN | MAX | PINION | GEAR | PINION | GEAR |
| 16.929000 | 16.934000 | 2.545000 | 2.490000 | 0.0200 | 0.0200 |

| ROOT DIA - GEAR | | FILLET RADIUS -MIN | | MAX UNDERCUT | | KO | KV |
|---|---|---|---|---|---|---|---|
| MIN | MAX | PINION | GEAR | PINION | GEAR | | |
| 16.180030 | 16.200000 | 0.030000 | 0.030000 | 0.0000 | 0.0000 | 1.0000 | 1.0000 |

—AT HPSTC IS LESS
IT OF 182000. PSI - INFINATE LIFE.

—AT HPSTC IS LESS
IT OF 182000. PSI - INFINATE LIFE.

### B E N D I N G   S T R E S S

| N | | GEAR |
|---|---|---|
| 32696.3555 | BSTC (HPSTC) | 125957.7188 |
| 68065.4707 | PP (STD) | 94310.6230 |
| 68065.4707 | PP (OP) | 94310.6230 |
| 123804.7285 | ESTC (LPSTC) | 24721.3926 |

### B E N D I N G   S T R E S S   (AGMA)

| N | | GEAR |
|---|---|---|
| 31572.2649 | BSTC (HPSTC) | 121227.6045 |
| 65725.4014 | PP (STD) | 90768.9580 |
| 65725.4014 | PP (OP) | 90768.9580 |
| 119548.3613 | ESTC (LPSTC) | 23793.0264 |

TRESS

| | GEAR |
|---|---|
| | 10289.5746 |

**OUTPUT SHEET**

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# APPENDIX VI

## AGMA STANDARD 220.02

Following is a reprint of "Tentative AGMA Standard for Rating the Strength of Spur Gear Teeth," by permission of V. C. Sears, American Gear Manufacturers Association.

## FOREWORD

This standard is for rating the strength of spur gear teeth. It contains the following:

BASIC RATING FORMULA

This section enumerates the factors known to affect strength. Numerical values are presented for those factors which have been evaluated by analytical means, test results or field experience. Suggestions are made for the factors which are not now capable of being expressed accurately. New knowledge and more definite measurement of these parameters will continually necessitate revisions and improvements.

In addition to the above, it is contemplated to publish design practices, such as AGMA 220.02A, having specific application under the heading of:

DESIGN PRACTICES FOR SPECIALIZED APPLICATIONS

It is recognized that it is sometimes desirable to provide simplified design practice data applicable to a specialized field of application. These individual design practices will enable enclosed speed reducer, mill gear, aircraft or other specialized product designers to record the modifications and limitations they wish to use.

Basic data illustrating the coordination of rating for all types of gears is contained in Tentative Information Sheet AGMA 225.01, "Strength of Spur, Helical, Herringbone and Bevel Gear Teeth.".

The first draft of the revision to this standard was prepared by the committee in September, 1955. It was approved by the AGMA membership as of April 7, 1963.

*Tables or other self-supporting sections may be quoted or extracted in their entirety. Credit lines should read: "Extracted from AGMA Standard for Rating the Strength of Spur Gear Teeth (AGMA 220.02), with the permission of the publisher, the American Gear Manufacturers Association, One Thomas Circle, Washington, D. C. 20005".*

# Personnel of
# Gear Rating Committee
# Technical Division
# January, 1964

E. J. Wellauer, Chairman, The Falk Corp., Milwaukee, Wis.

D. L. Borden, The Falk Corp., Milwaukee, Wis.

W. Coleman, Gleason Works, Rochester, New York

D. W. Dudley, General Electric Co., West Lynn, Mass.

J. H. Glover, Ford Motor Co., Dearborn, Michigan

I. Koenig, Hewitt-Robins, Inc., Chicago, Illinois

C. F. Schwan, Reliance Electric & Engineering Co., Cleveland, Ohio

J. C. Straub, Wheelabrator Corp., Mishawaka, Indiana

F. A. Thoma, De Laval Turbine, Inc., Trenton, New Jersey

N. A. Wilson, Morgan Construction Co., Worcester, Mass.

G. L. Scott, AGMA, Washington, D. C.

# TENTATIVE
# AGMA STANDARD
# STRENGTH OF SPUR GEAR TEETH

## Basic Rating Formula

## 1. Scope

**1.1** This standard presents the fundamental formulas for the strength of spur gear teeth. It includes all of the factors which are known to affect gear tooth strength. This standard is based on Information Sheet AGMA 225.01 and is therefore coordinated with strength ratings for helical and bevel gears.

**1.2** Both pinion and gear teeth must be checked for bending strength rating to account for differences in geometry factors, material properties, and numbers of tooth contact cycles under load.

**1.3** Other AGMA standards contain numerical values to be used to rate gears for specific applications. These should be consulted when applicable.

**1.4** Where no applicable specific AGMA standard is established, numerical values may be estimated for the factors in the fundamental formula and an approximate strength rating computed.

**1.5** The formulas presented in this reference apply to external gears unless otherwise noted.

**1.6** The symbols used, wherever applicable, conform to Standard AGMA 111.03 "Letter Symbols for Gear Engineering" (ASA B6.5-1954) and "Letter Symbols for Mechanics of Solid Bodies" (ASA Z10.3-1948).

## 2. Fundamental Bending Stress Formula

**2.1** The basic equation for the bending stress in a gear is calculated as follows:

$$s_t = \frac{W_t K_o}{K_v} \frac{P_d}{F} \frac{K_s K_m}{J}$$

Where:

$s_t$ = calculated tensile bending stress at the root of the tooth, psi

Load
- $W_t$ = transmitted tangential load at operating pitch dia. lbs. (see Section 4).
- $K_o$ = overload factor (see Section 9)
- $K_v$ = dynamic factor (see Section 8)

Tooth Size
- $P_d$ = diametral pitch
- $F$ = face width, in.

Stress Distribution
- $K_s$ = size factor (see Section 7)
- $K_m$ = Load distribution factor (see Section 6)
- $J$ = geometry factor (see Section 5)

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# AGMA STANDARD
# STRENGTH OF SPUR GEAR TEETH

**2.1.1** Note that the above equation is divided into three groups of terms, the first of which is concerned with the load, the second with tooth size, and the third with stress distribution.

**2.2** The relation of calculated stress to allowable stress is:

$$s_t \leq \frac{s_{at} K_L}{K_R K_T}$$

Where:

$s_{at}$ = allowable bending stress for material, psi (see Section 13)

$s_t$ = calculated bending stress, psi (see paragraph 2.1)

$K_L$ = life factor (see Section 11)

$K_T$ = temperature factor (see Section 12)

$K_R$ = factor of safety (see Section 10)

### 3. Fundamental Power Formula

**3.1** In preparing handbook data, for gear designs already developed, the following formula can be used to directly calculate the power which can be transmitted by a given gear set:

$$P_{at} = \frac{n_P \, d \, K_v}{126,000 \, K_o} \frac{F}{K_m} \frac{J}{K_s P_d} \frac{s_{at} K_L}{K_R K_T}$$

Where:

$P_{at}$ = allowable power of gear set, hp

$n_P$ = pinion speed, rpm

$d$ = operating pitch diameter of pinion, in.

### 4. Transmitted Tangential Load

**4.1** The transmitted tangential load is calculated directly from the power transmitted by the gear set. (When operating near a critical speed of the drive, a careful analysis of conditions must be made.) When the transmitted load is not uniform, consideration should be given not only to the peak load and its anticipated number of cycles, but also to intermediate loads and their number of cycles.

**4.2** The transmitted tangential load is:

$$W_t = \frac{33,000 \, P}{v_t} = \frac{2T}{d} = \frac{126,000 \, P}{n_P d}$$

Where:

$P$ = power transmitted, hp

$T$ = pinion torque, lb.in.

$v_t$ = pitch line velocity, fpm

### 5. Geometry Factor — $J$

**5.1** The geometry factor evaluates the shape of the tooth, the position at which the most damaging load is applied, stress concentration due to geometric shape and the sharing of load.

**5.2** See Appendix A for a further discussion of spur gear geometry factors, and paper AGMA 229.07, "Spur and Helical Gear Geometry Factors."

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5.3 Accurate spur gears develop the most critical stress when load is applied at the highest point of the tooth where a single pair of teeth is carrying all the load. Less accurate spur gears, having errors that prevent two pairs of teeth from sharing the load, may be stressed most heavily when load is applied at the tip. Figures 1A and 1B show the geometry factor for equal addendum involute spur gears of 20 deg and 25 deg pressure angle. In these curves, it is assumed that the theoretical stress concentration factor is not affected seriously by surface finish, plasticity, residual stresses or other factors.

5.3.1 Table 1 shows the variation in base pitch between the gear and pinion which determines whether or not load sharing exists in 20 degree pressure angle spur gears.

## 6. Load Distribution Factor — $K_m$

6.1 The load distribution factor depends upon the combined effect of:

1. misalignment of axes of rotation

2. lead deviations

3. elastic deflection of shafts, bearings and housing.

6.2 Figures 2 and 3 illustrate misalignment and its effect on load distribution.

6.3 The effect of different rates of spur gear misalignment is shown in Figure 4.

6.4 When the misalignment is known, use Figure 4 to select $K_m$. $F_m$ represents the face width having just 100 per cent contact for a given tangential load and alignment error. Generally $F_m$ should exceed $F$.

6.5 Manufacturers of precision gears with face widths greater than 6 inches generally find it necessary to control misalignment by other means than allowed rates of misalignment. To handle such cases, Table 2 shows appropriate values of $K_m$.

6.6 When the estimated or actual misalignment is not known, the $K_m$ factor may be obtained from Table 3.

## 7. Size Factor — $K_s$

7.1 The size factor reflects non-uniformity of material properties. It depends primarily on:

1) tooth size;

2) diameter of parts;

3) ratio tooth size to diameter of part;

4) face width;

5) area of stress pattern;

6) ratio of case depth to tooth size;

7) hardenability and heat treatment of materials.

7.2 The size factor may be taken as unity for most spur gears provided a proper choice of steel is made for the size of the parts and the case depth or hardness pattern is adequate.

7.3 Standard size factors for spur gear teeth have not yet been established for cases where there is a detrimental size effect. In such cases a size factor greater than unity should be used.

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# AGMA STANDARD
# STRENGTH OF SPUR GEAR TEETH

### Table 1 Limiting Error in Action for Steel Spur Gear

### (Variation in Base Pitch)

| Number of Pinion Teeth | Allowable Error When Teeth Share Load* | | | | | Amount of Error When Teeth Fail to Share Load** | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Load Per In. of Face | | | | | Load Per In. of Face | | | | |
| | 500 lb. | 1,000 lb. | 2,000 lb. | 4,000 lb. | 8,000 lb. | 500 lb. | 1,000 lb. | 2,000 lb. | 4,000 lb. | 8,000 lb. |
| 15 | 0.0004 | 0.0007 | 0.0014 | 0.0024 | 0.0042 | 0.0006 | 0.0011 | 0.0023 | 0.0039 | 0.0064 |
| 20 | 0.0003 | 0.0006 | 0.0011 | 0.0020 | 0.0036 | 0.0006 | 0.0011 | 0.0023 | 0.0039 | 0.0064 |
| 25 | 0.0002 | 0.0005 | 0.0009 | 0.0017 | 0.0030 | 0.0006 | 0.0011 | 0.0023 | 0.0039 | 0.0064 |

*Use upper curves on Fig. 1 — highest point of single tooth loading.

**Use lower curve of Fig. 1 — tip loading.

### 8. Dynamic Factor — $K_v$

8.1 The dynamic factor depends on:

1) effect of tooth spacing and profile errors.

2) effect of pitch line and rotational speeds.

3) inertia and stiffness of all rotating elements.

4) transmitted load per inch of face.

5) tooth stiffness.

8.2 Figure 5 shows some of the dynamic factors that are commonly used.

Curve No. 1 — To be used with high precision shaved or ground spur gears where the effect of the items listed in paragraph 8.1 are such that no appreciable dynamic load is developed.

Curve No. 2 — To be used with high precision shaved or ground spur gears when the items listed in paragraph 8.1 can develop a dynamic load.

Curve No. 3 — To be used with spur gears finished by hobbing or shaping.

8.3 When milling cutters are used to cut the teeth or inaccurate teeth are generated, lower dynamic factors than shown must be used since the dynamic factor reflects the effect of inaccuracies in profile, tooth spacing and runout.

### 9. Overload Factor — $K_o$

9.1 The overload factor makes allowances for the roughness or smoothness of operation of both the driving and driven apparatus. Specific overload factors can only be established after considerable field experience is gained in a particular application.

9.2 In determining the overload factor, consideration should be given to the fact that many prime movers develop momentary overload torques appreciably greater than those determined by the nameplate ratings of either the prime mover or the driven apparatus.

9.3 In the absence of specific overload factors, the values in Table 4 should be used.

# AGMA STANDARD
# STRENGTH OF SPUR GEAR TEETH

### Table 2   Load Distribution Factor for Precision Wide-Face Spur Gears — $K_m$

| Ratio of $\frac{F}{d}$ | Contact | Load Distribution Factor — $K_m$ |
|---|---|---|
| | 95% face width contact obtained at 1/3 torque | 1.4 at 1/3 torque |
| | 95% face width contact obtained at full torque | 1.1 at full torque |
| | 75% face width contact obtained at 1/3 torque | 1.8 at 1/3 torque |
| | 95% face width contact obtained at full torque | 1.3 at full torque |
| 1.0 or less | 35% face width contact obtained at 1/3 torque | 2.5 at 1/3 torque |
| | 95% face width contact obtained at full torque | 1.9 at full torque |
| | 20% face width contact obtained at full torque | 4.0 at 1/3 torque |
| | 75% face width contact obtained at full torque | 2.5 at full torque |
| | Teeth are crowned | |
| | 35% face width contact at 1/3 torque | 2.5 at 1/3 torque |
| | 85% face width contact at full torque | 1.7 at full torque |
| | Calculated combined twist and bending of pinion not over .001 in. over entire face | |
| | Pinion not over 250 Bhn hardness | |
| | 75% contact obtained at 1/3 torque | 2.0 at 1/3 torque |
| | 95% contact obtained at full torque | 1.4 at full torque |
| over 1 less than 2 | Calculated combined twist and bending of pinion not over .0007 in. over entire face | |
| | Pinion not over 350 Bhn hardness | |
| | 75% contact obtained at 1/3 torque | 2.0 at 1/3 torque |
| | 95% contact obtained at full torque | 1.4 at full torque |
| | 30% contact obtained at 1/3 torque | 4.0 at 1/3 torque |
| | 75% contact obtained at full torque | 3.0 at full torque |

# AGMA STANDARD
# STRENGTH OF SPUR GEAR TEETH

**Table 3   Load Distribution Factor — $K_m$**

| Condition of Support | Face Width, in. | | | |
|---|---|---|---|---|
| | 2 in. Face and Under | 6 in. Face | 9 in. Face | 16 in. Face and Over |
| Accurate mountings, low bearing clearances, minimum elastic deflection, precision gears | 1.3 | 1.4 | 1.5 | 1.8 |
| Less rigid mountings, less accurate gears, contact across full face | 1.6 | 1.7 | 1.8 | 2.0 |
| Accuracy and mounting such that less than full face contact exists | over 2.0 | | | |

**Table 4   Overload Factors* — $K_o$**

| Power Source | Load on Driven Machine | | |
|---|---|---|---|
| | Uniform | Moderate Shock | Heavy Shock |
| Uniform | 1.00 | 1.25 | 1.75 or higher |
| Light Shock | 1.25 | 1.50 | 2.00 or higher |
| Medium Shock | 1.50 | 1.75 | 2.25 or higher |

* Note that this table is for speed decreasing drives only. For speed increasing drives add

$$0.01 \left( \frac{N_G}{N_P} \right)^2 \text{ to the factors in Table 4.}$$

Where:

$N_P$ = number of teeth in the pinion

$N_G$ = number of teeth in the gear.

**9.4** Service factors have been established where field data is available for specific applications. These service factors include not only the overload factor, but also the life factor and factor of safety. Service factors for many applications are listed in other AGMA Standards, and should be used whenever available. If a specific service factor is used in place of the overload factor $K_o$, use a value of 1.0 for $K_R$ and $K_L$.

## 10. Factor of Safety — $K_R$

**10.1** The factor of safety is introduced in this equation to offer the designer an opportunity to design for high reliability or, in some instances, to design for a calculated risk. Table 5 shows a suggested list of factors of safety to be applied to the fatigue strength of the material rather than to the tensile strength. For this reason, the values are much smaller than customarily used in other branches of machine design.

**10.1.1** Failure in the following table does not mean an immediate failure under applied load, but rather a shorter life than the minimum specified.

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# AGMA STANDARD
# STRENGTH OF SPUR GEAR TEETH

**Table 5  Factors of Safety — $K_R$**

**Fatigue Strength**

| Requirements of Application | $K_R$ |
|---|---|
| High Reliability | 1.50 or higher |
| Fewer than 1 failure in 100 | 1.00 |
| Fewer than 1 failure in 3 | 0.70 |

**10.2** Table 6 shows safety factors to be applied to the yield strength of the material.  These values must be applied to the maximum peak load to which the gears are subjected.

**Table 7  Life Factor — $K_L$**

| Number of Cycles | $K_L$ | | | |
|---|---|---|---|---|
| | 160 Bhn | 250 Bhn | 450 Bhn | case carb.* |
| Up to 1,000 | 1.6 | 2.4 | 3.4 | 2.7 |
| 10,000 | 1.4 | 1.9 | 2.4 | 2.0 |
| 100,000 | 1.2 | 1.4 | 1.7 | 1.5 |
| 1 million | 1.1 | 1.1 | 1.2 | 1.1 |
| 10 million and over | 1.0 | 1.0 | 1.0 | 1.0 |

* case carburized 55-63 $R_c$

**Table 6  Factors of Safety — $K_R$**

**Yield Strength**

| Requirements of Application | $K_R$ |
|---|---|
| High Reliability | 3.00 or higher |
| Industrial | 1.33 |

## 11.  Life Factor — $K_L$

**11.1** The life factor adjusts the allowable loading for the required number of cycles.  Table 7 shows typical values, for use with the allowable stress values of Figure 6 or Table 8.

## 12.  Temperature Factor — $K_T$

**12.1** When gears operate at oil or gear blank temperatures not exceeding 250 degree F, $K_T$ is generally taken as unity.  In some instances, it is necessary to use a $K_T$ value greater than unity for case carburized gears at a temperature above 160 degree F.  One basis of correction is:

$$K_T = \frac{460 + T_F}{620}$$

Where:

$T_F$ = The peak operating oil temperature in degrees Fahrenheit.

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### 13. Allowable Bending Stress —

$$s_{at} \text{ and } s_{ay}$$

**13.1** An allowable design bending stress for unity application factor and 10 million cycles of load application is determined by field experience, for each material and condition of that material. This stress is designated $s_{at}$.

**13.2** The allowable stress for gear materials varies considerably with heat treatment, forging or casting practice, material composition, and with various surface treatments.

**13.3** Frequently, shot peening permits a higher allowable stress to be used.

**13.4** The allowable fatigue design stress for steel is shown in Figure 6. These values are suggested for general design purposes.

**13.5** The allowable fatigue design stress for surface hardened steel and other materials is shown in Table 8.

**13.6** Use 70 per cent of the $s_{at}$ values for idler gears and other gears where the teeth are loaded in both directions.

**13.7** When the gear is subjected to infrequent momentary high overloads the maximum allowable stress is determined by the allowable yield properties rather than the fatigue strength of the material. This stress is designated as $s_{ay}$. Figure 7 shows suggested values for allowable yield strength, for through hardened steel. In these cases the design should be checked to make certain that the teeth are not permanently deformed. When yield is the governing stress, the stress concentration factor is sometimes considered ineffective.

**Table 8  Allowable Fatigue Design Stress — $s_{at}$**

| Material | Material Hardness, min. | $s_{at}$-psi |
|---|---|---|
| Steel | | |
| Case Carburized and Hardened | 55 $R_c$ | 55-65,000 |
| Induction or Flame Hardened | | |
| Hard Root | 300 Bhn | use values from Fig. 6 |
| Unhardened Root | — | 22,000 |
| Cast Iron | | |
| AGMA Grade 20 | — | 5,000 |
| AGMA Grade 30 | 175 Bhn | 8,500 |
| AGMA Grade 40 | 200 Bhn | 13,000 |
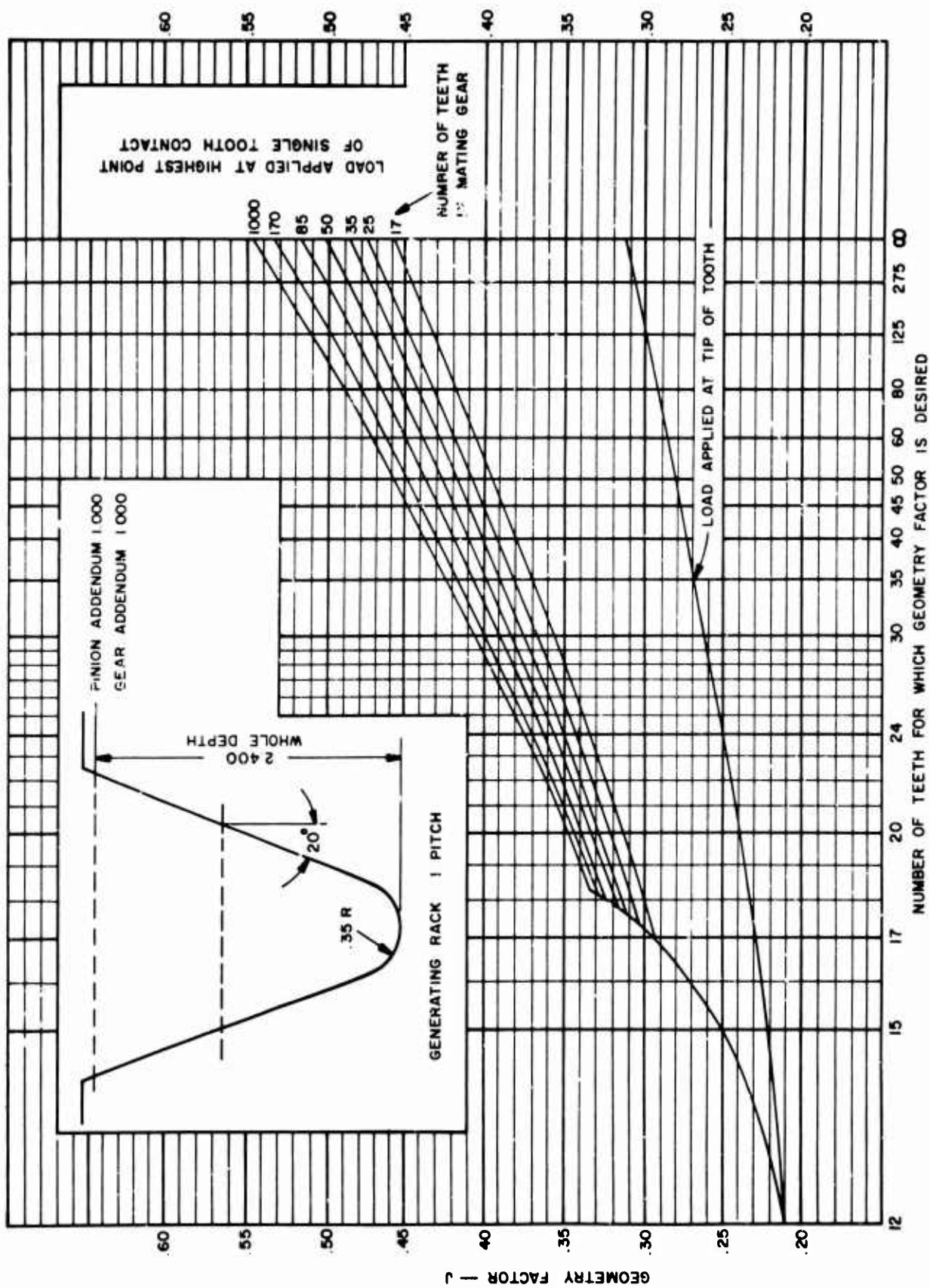
FIG. IA GEOMETRY FACTORS — 20° SPUR — STANDARD ADDENDUM

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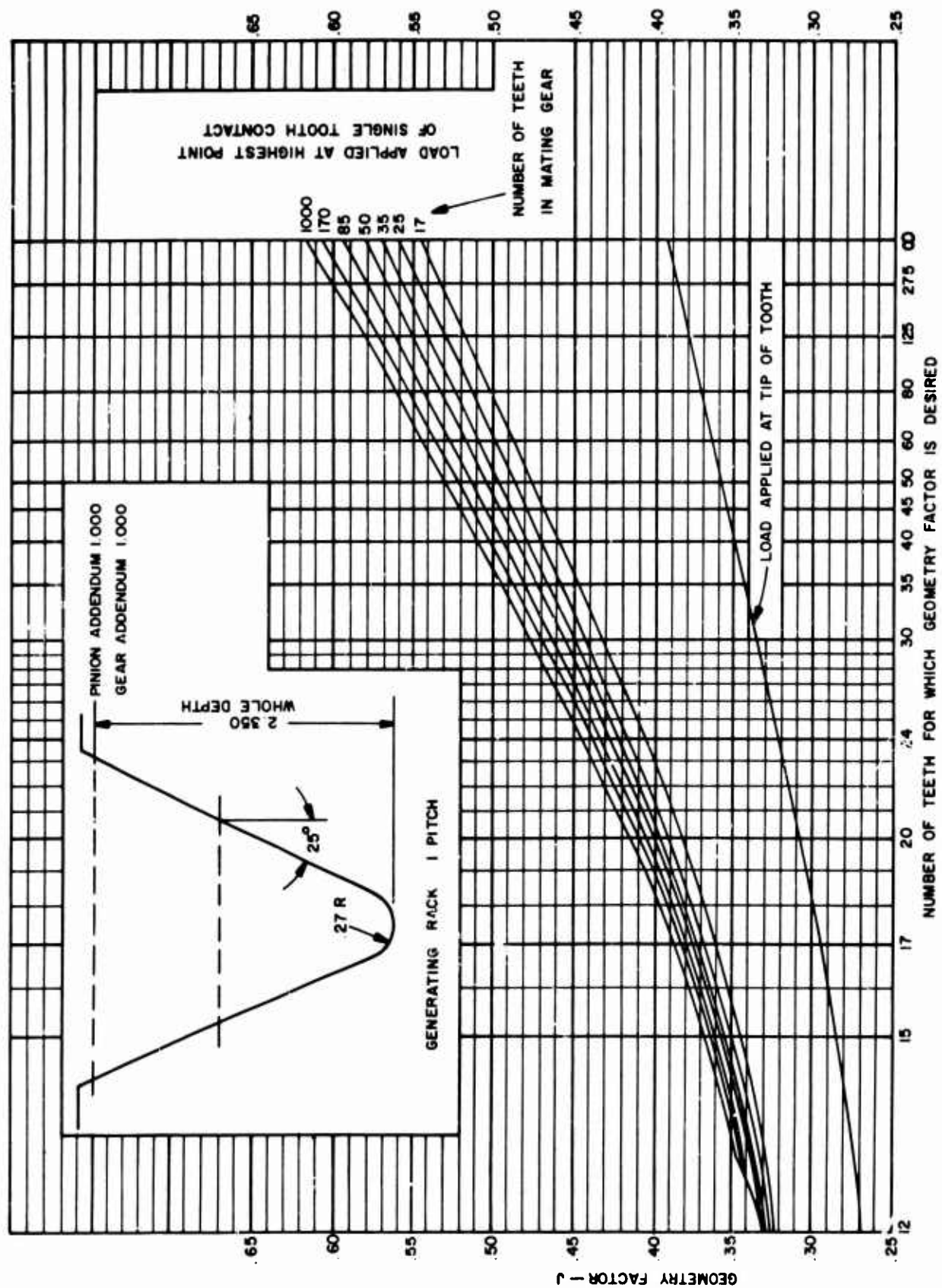
FIG. IB  GEOMETRY FACTORS — 25° SPUR — STANDARD ADDENDUM

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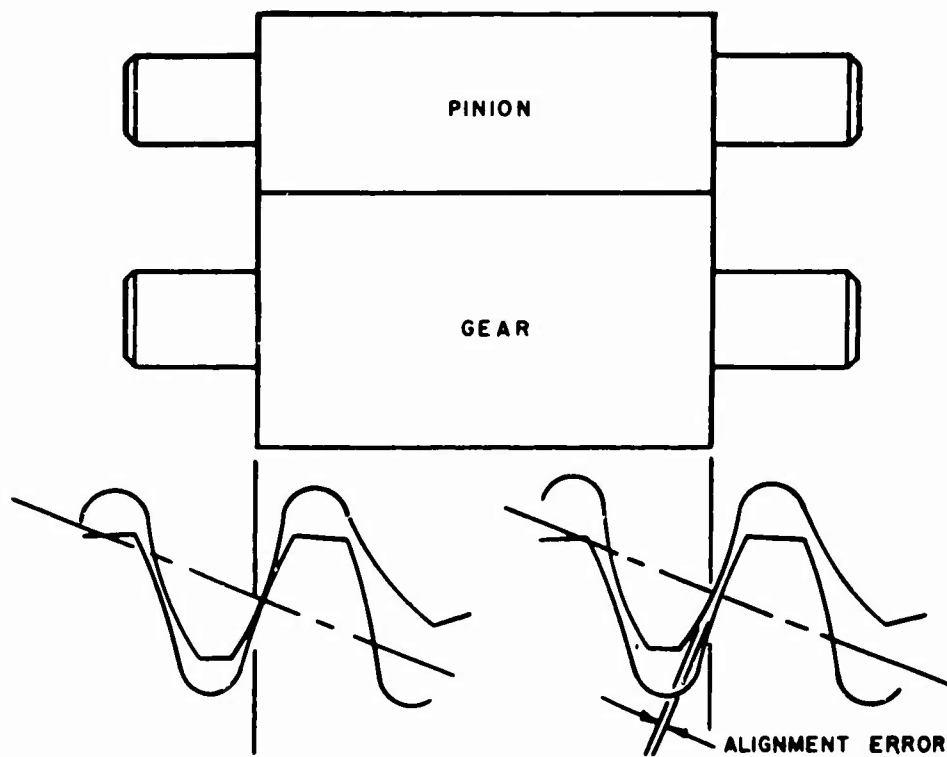
FIG. 2   EXAMPLE OF A PINION AND GEAR MISALIGNED UNDER
NO LOAD.  TEETH CONTACT AT LEFT HAND END
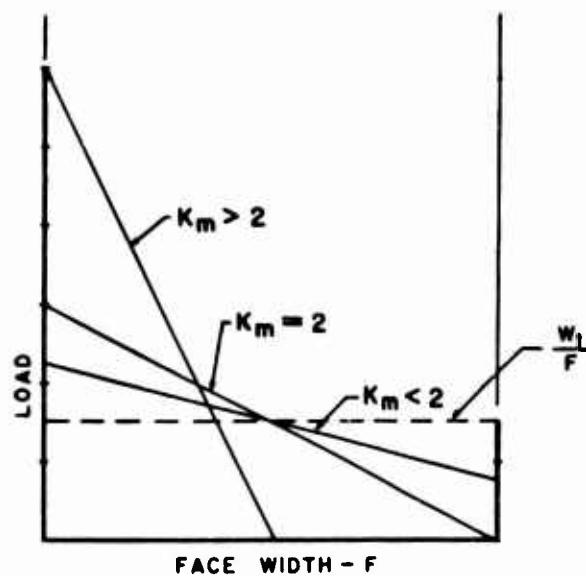AND ARE OPEN AT RIGHT HAND END.



FIG. 3   LOAD DISTRIBUTION ACROSS FACE WIDTH
FOR VARIOUS CONTACT CONDITIONS

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$$C_e = \frac{W_t}{1000\,e}$$

$W_t$ = TANGENTIAL LOAD - LBS

$e$ = ALIGNMENT ERROR INCHES/INCH

FORCE - ERROR COEFFICIENT - $C_e$

$F_m$

$C_m$

$\frac{F_m}{F}$

F = FACE WIDTH (INCHES)
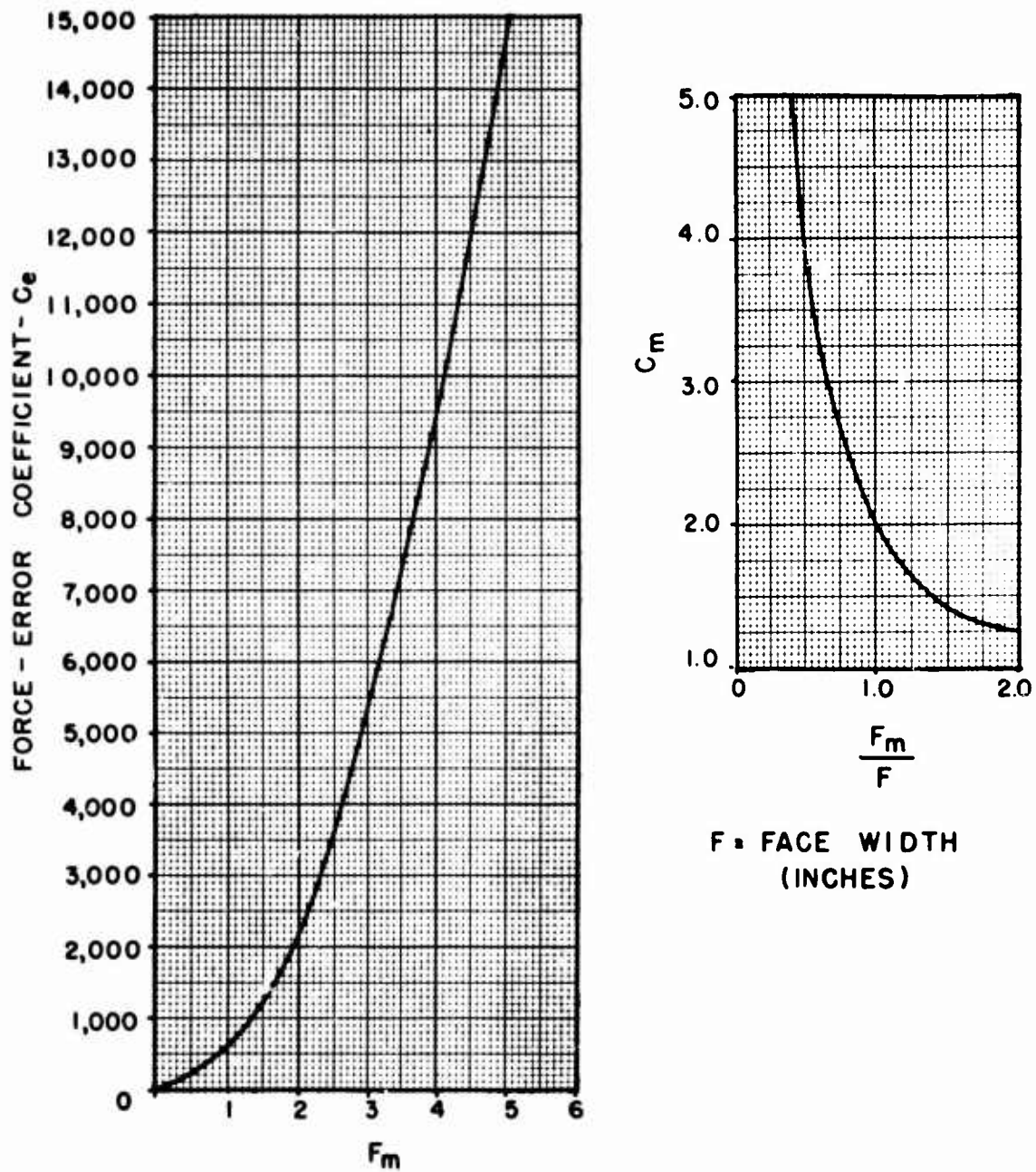
FIG. 4   SPUR GEAR LOAD DISTRIBUTION FACTOR — $K_m$

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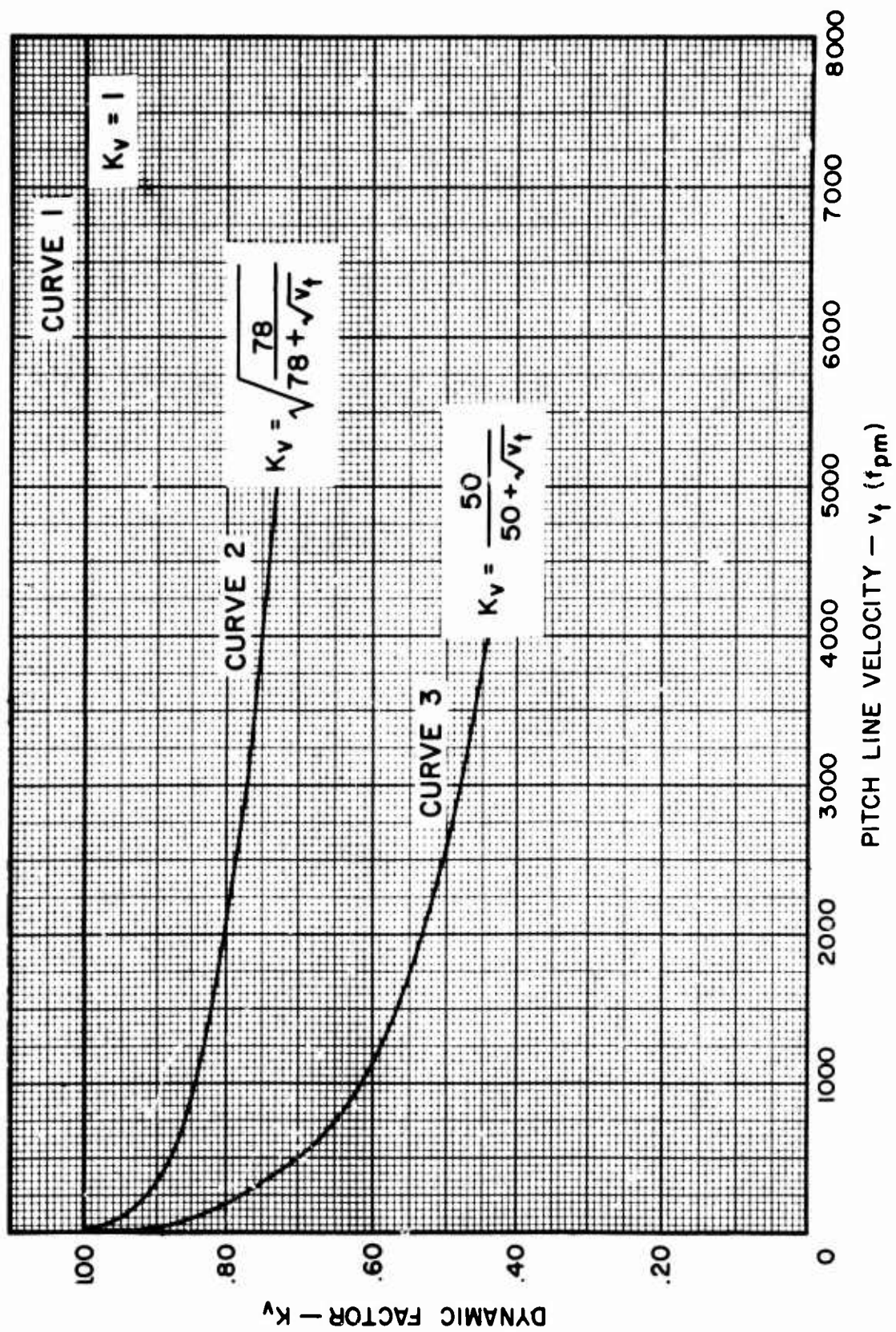
FIG. 5   DYNAMIC FACTOR — $K_v$
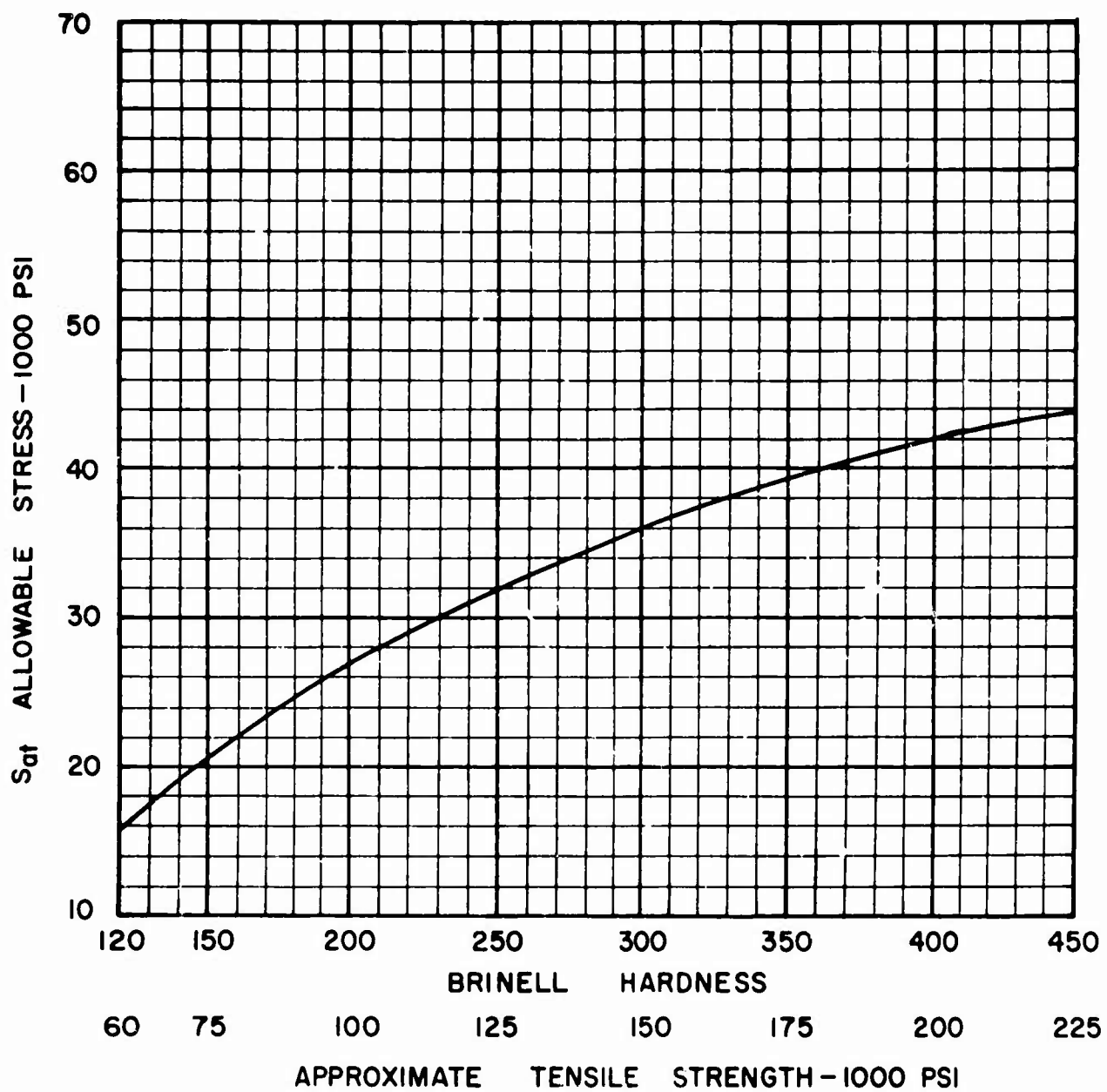
FIG. 6   ALLOWABLE   FATIGUE   STRESS   FOR   STEEL   GEARS—$S_{at}$

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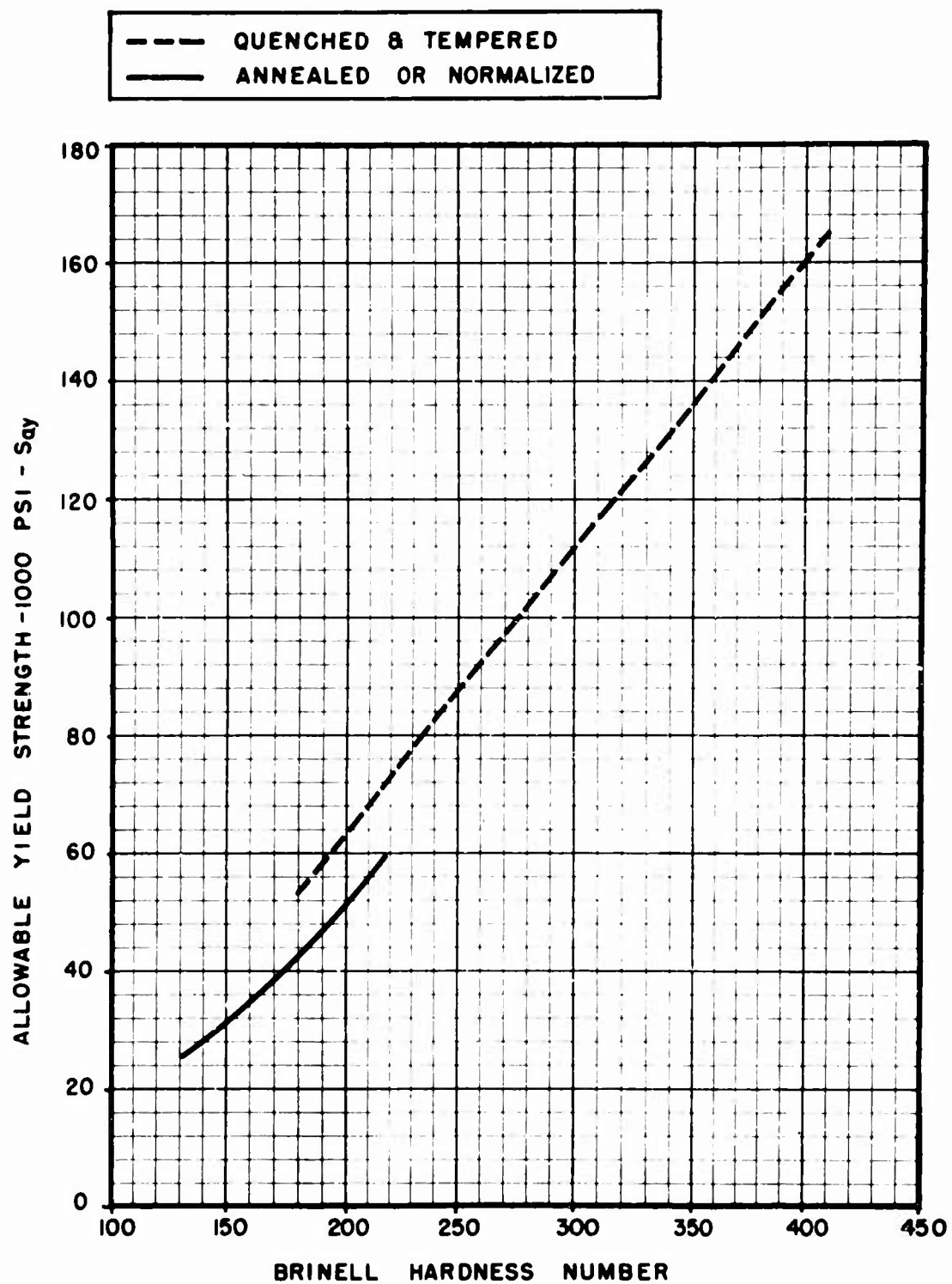
FIG. 7    ALLOWABLE  YIELD  STRENGTH – $S_{ay}$

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# APPENDIX A

## SPUR GEAR GEOMETRY FACTOR

### 1. Geometry Factor — $J$

$$J = \frac{Y}{K_f \, m_N}$$

Where:

$J$ = geometry factor

$Y$ = tooth form factor

$K_f$ = stress correction factor

$m_N$ = load sharing ratio

### 2. Tooth Form Factor — $Y$

2.1 $Y$ is determined for the most critical position of load application. This is at the tip of the tooth when load sharing does not exist and usually at the highest load position for single tooth contact when load sharing does exist.

2.2 The $Y$ factor, which considers both the tangential (bending) and radial (compressive) components of the load is calculated as follows:

$$Y = \frac{1}{\dfrac{\cos \phi_L}{\cos \phi} \left( \dfrac{1.5}{X} - \dfrac{\tan \phi_L}{t} \right)}$$

Where:

$\phi$ = pressure angle

2.3 Use the following procedure to determine $Y$.

2.3.1 Lay out a generated tooth profile at a scale of one diametral pitch $(P_d)$, as shown in Figures A1 and A2.

2.3.2 When load sharing exists (Fig. A1), lay a scale tangent to the base circle and locate the position where the distance from the intersection point with the pitch circle to the intersection point with the profile equals distance $z_c$ — inches (obtained from Figures A3 or A4). This locates line $aa$.

2.3.3 When load sharing does not exist (Fig. A2), draw line $aa$ through point $p$ and tangent to the base circle. This locates line $aa$.

2.3.4 Through point $f$ draw line $bb$ perpendicular to the tooth center line. The included angle between lines $aa$ and $bb$ is angle $\phi_L$.

2.3.5 Draw line $cde$ tangent to the tooth fillet radius $(r_f)$ at $e$, intersecting line $bb$ at $d$ and the tooth center line at $c$ so that $cd = de$.

2.3.6 Draw line $fe$.

2.3.7 Through point $e$ draw a line perpendicular to $fe$, intersecting the tooth center line at $n$.

2.3.8 Through point $e$, draw a line $me$ perpendicular to the tooth center line.

2.3.9 Measure the following from the tooth layout:

$mn = X$ — inches

$me = t/2$ — inches

angle $\phi_L$

2.3.10 Calculate form factor $Y$.

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# APPENDIX A

## 3. Stress Correction Factor — $K_f$

### 3.1 Stress correction factor depends on:

1) effective stress concentration;

2) location of load;

3) plasticity effects;

4) residual stress effects;

5) material composition effects;

6) surface finish:

   a) resulting from gear production

   b) resulting from service.

7) Hertz stress effects;

8) size effect;

9) end of tooth effects.

3.2 The following stress correction factor is that of Dolan and Broghamer and only includes the effects of items 1 and 2.

$$K_f = H + \left(\frac{t}{r_f}\right)^J \left(\frac{t}{h}\right)^L$$

Where:

H, J and L are obtained from Table A-1. For other pressure angles, the values of H, J and L can be obtained by interpolation and extrapolation.

Table A-1 Values for H, J and L

| Pressure Angle | H | J | L |
|---|---|---|---|
| $14\frac{1}{2}°$ | 0.22 | 0.20 | 0.40 |
| 20° | 0.18 | 0.15 | 0.45 |
| 25° | 0.14 | 0.11 | 0.50 |

$h$ = distance $fm$ measured from the layout — inches

$\frac{t}{2}$ = distance $me$ measured from the layout — inches

$r_f = r_1 + r_T$

Where:

$r_T$ = edge radius of tool — inches. For a cutter with chamfered teeth, take $r_T = 0$.

$$r_1 = \frac{b_1^2}{R_o + b_1}$$

Where:

$R_o$ = the relative radius of curvature of the pitch circle of the gear and the pitch line or pitch circle of the generating tool. For generation by a rack or hob, $R_o$ equals the pitch radius R of the gear being generated. For generation by a pinion-shaped cutter, $1/R_o = 1/R + 1/R_c$, where $R_c$ is the pitch radius of the cutter.

$$b_1 = b - r_T$$

Where:

$b$ = dedendum — inches

3.3 Plasticity reduces the effect of stress concentration and is partially measured by the life factor of Table 7. When more accurate data such as notch sensitivity values are available, they may be used.
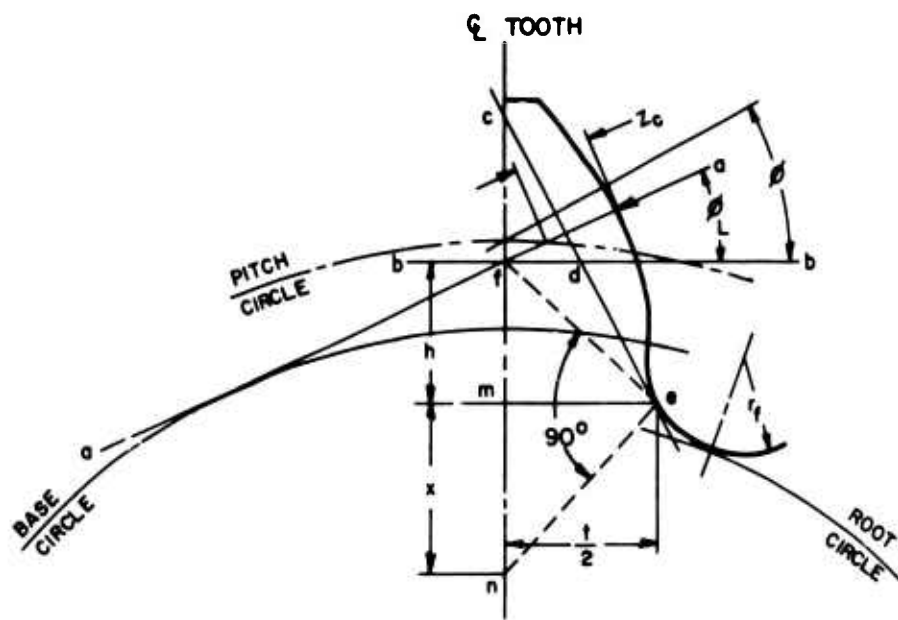
3.4 If more exact values for the stress correction factor are available, they may be used.
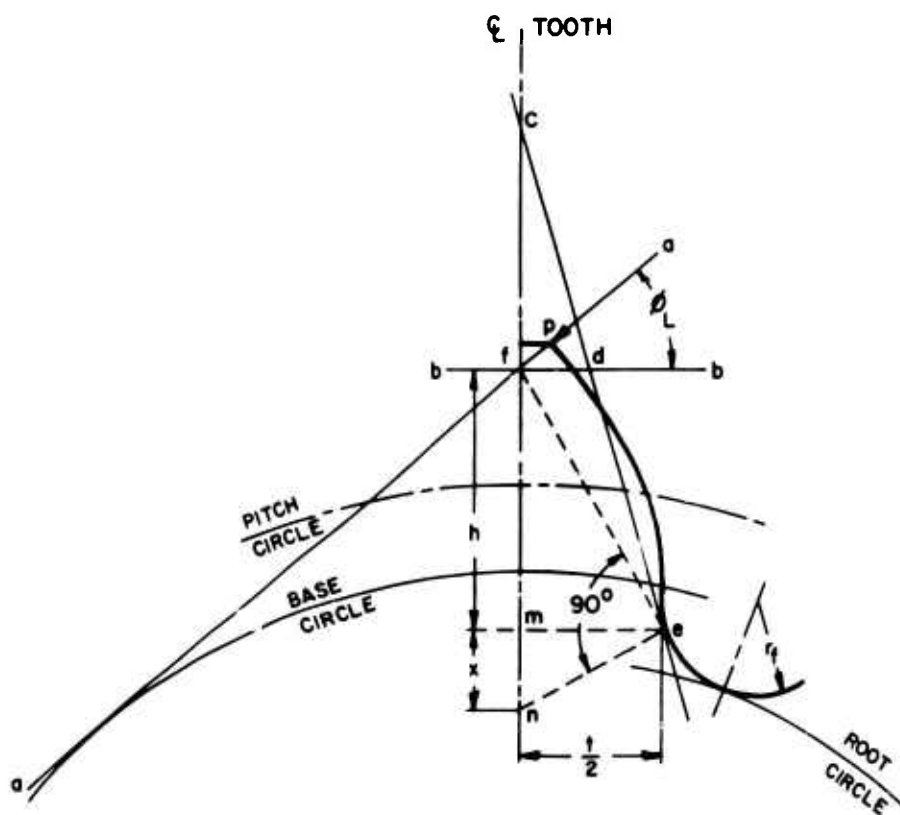
## 4. Load Sharing Ratio — $m_N$

4.1 Load sharing ratio is influenced by profile contact ratio.

4.2 The most critical position of spur gear load application normally occurs when only one tooth is in contact.
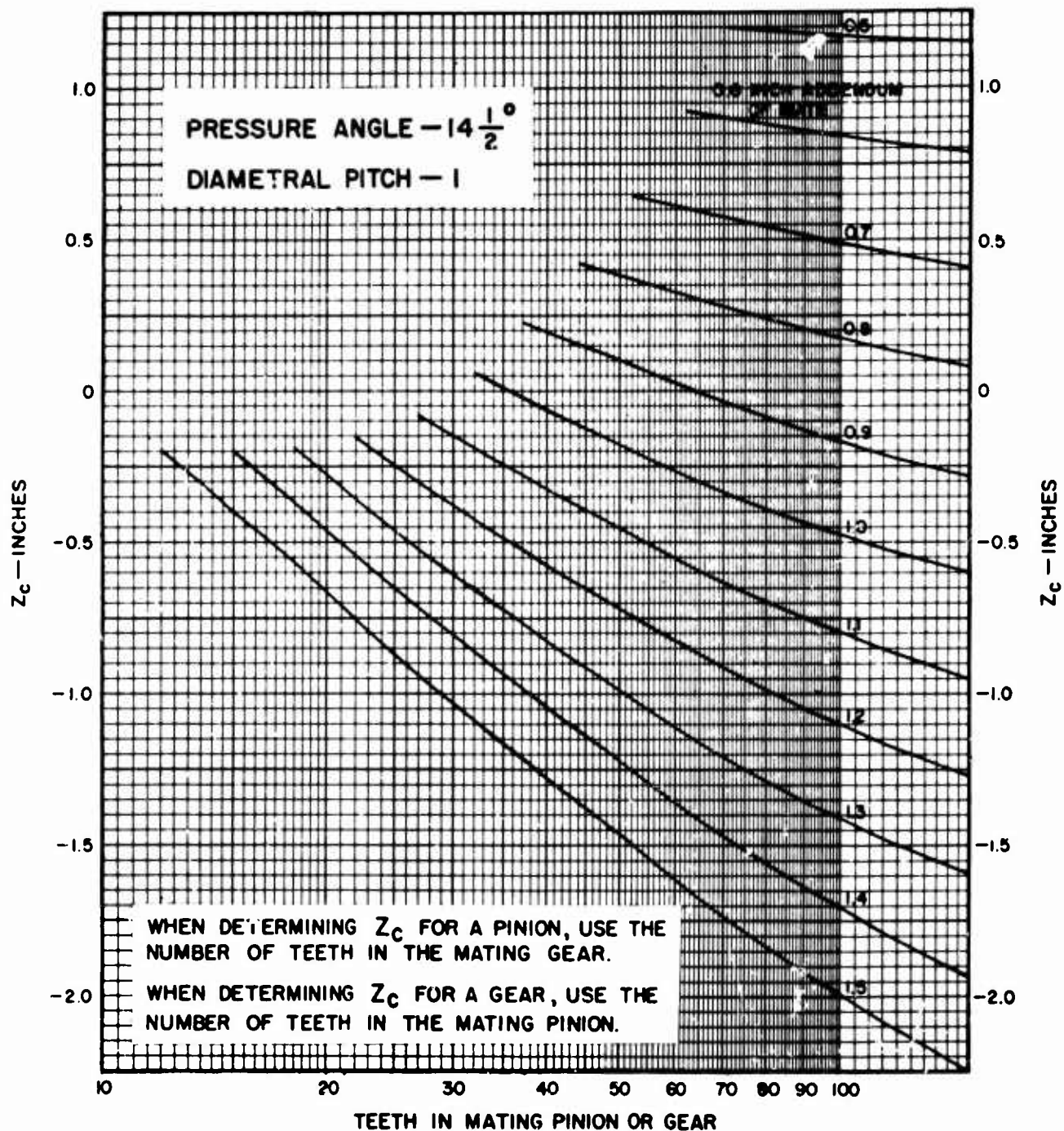
Therefore, $m_N = 1.0$.

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LOAD AT HIGHEST POINT FOR SINGLE TOOTH CONTACT

FIG. A-I TOOTH FORM FACTOR LAYOUT WITH LOAD SHARING



TIP LOADING

FIG. A-2 TOOTH FORM FACTOR LAYOUT WITHOUT LOAD SHARING

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FIG. A-3  $Z_c$ — FOR HIGHEST POINT OF SINGLE TOOTH CONTACT
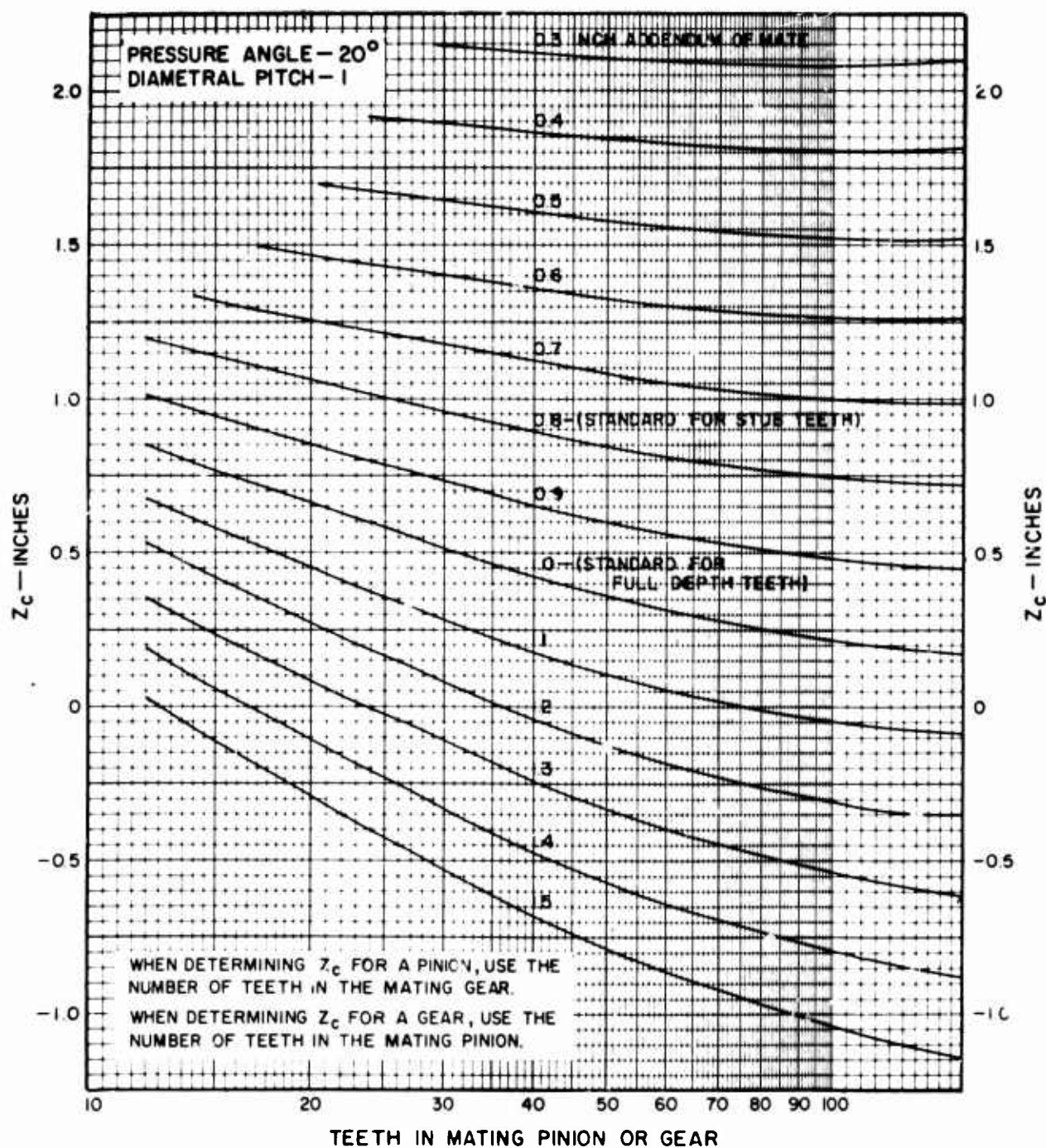WHEN LOAD SHARING EXISTS BETWEEN TEETH

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FIG. A-4  $Z_c$ — FOR HIGHEST POINT OF SINGLE TOOTH CONTACT
WHEN LOAD SHARING EXISTS BETWEEN TEETH

The user of this Standard (AGMA 220.02) may find these other AGMA Standards of value as reference data:

| Number | Title |
|--------|-------|
| AGMA 110.03 | Gear Tooth Wear and Failure |
| AGMA 112.03 | Terms, Definitions, and Illustrations |
| AGMA 115.01 | Basic Gear Geometry — Reference Information |
| AGMA 201.02 | Tooth Proportions for Coarse-Pitch Involute Spur Gears |
| AGMA 207.04 | 20-Degree Involute Fine-Pitch System for Spur and Helical Gears |
| AGMA 208.02 | System for Straight Bevel Gears |
| AGMA 212.02 | Surface Durability (Pitting) Formulas for Straight Bevel and Zerol Bevel Gear Teeth |
| AGMA 216.01 | Surface Durability (Pitting) Formulas for Spiral Bevel Gear Teeth |
| AGMA 221.02 | Strength of Helical and Herringbone Gear Teeth |
| AGMA 225.01 | Strength of Spur, Helical, Herringbone and Bevel Gear Teeth |
| AGMA 241.02 | Gear Materials — Steel |
| AGMA 244.01 | Nodular Iron Gear Materials |
| AGMA 245.01 | Recommended Procedure for Cast Steel Gear Materials |
| AGMA 247.01 | Recommended Procedure for Steel, Nitriding, Materials and Process |
| AGMA 248.01 | Recommended Procedure for Induction Hardened Gears and Pinions |
| AGMA 249.01 | Recommended Procedure for Flame Hardening |
| AGMA 250.02 | Lubrication of Industrial Enclosed Gearing |
| AGMA 252.01 | Mild Extreme Pressure Lubricants for Industrial Enclosed Gearing |
| AGMA 390.01 | Gear Classification Manual for Spur, Helical and Herringbone Gears |
| AGMA 411.01 | Design Procedure for Aircraft Engine and Power Take-Off Spur Gears |

A more complete list of AGMA Standards published by the American Gear Manufacturers Association is available upon request.

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**13. ABSTRACT**

An analytical and experimental study to evaluate factors for accurate appraisal of gear tooth bending strength is described. A design analysis consisting of a thorough review of current methods of calculating bending strength is presented. Experimental evaluation of strain gage measured stresses, photostress patterns, and a dynamic test at high speed are also presented for substantiation of the theoretical stress analysis. Results from a designed static fatigue test are presented to evaluate the effect of four geometric variables, i.e., diametral pitch, pressure angle, fillet radius size, and fillet configuration. A final computer program is described to calculate bending stress as substantiated by design analysis, experimental evaluation, and static fatigue test. It is shown that the thorough consideration of geometric variables permits a more precise assessment of bending strength for life expectance of lightweight aircraft gearing.

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